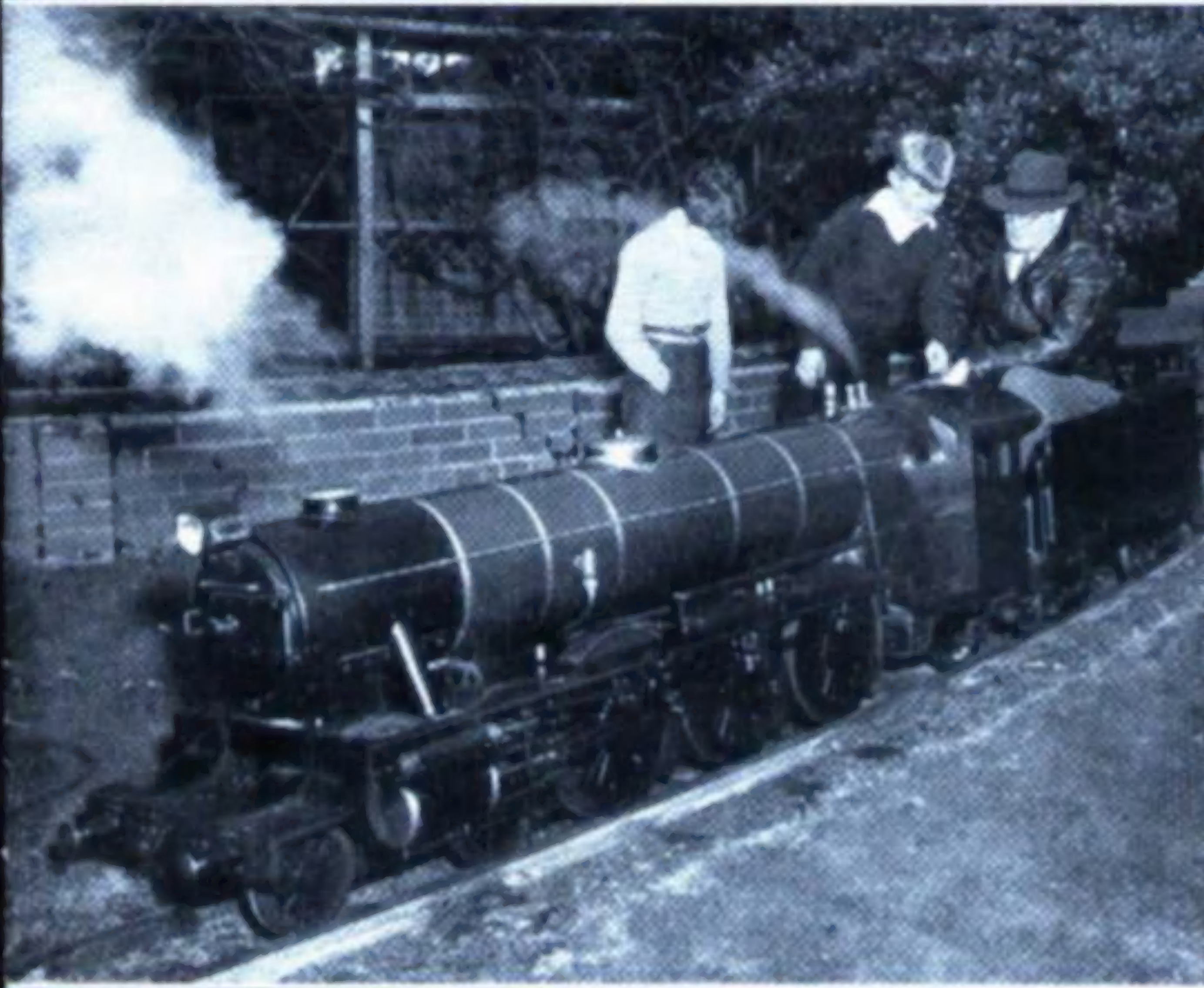
THE MODEL ENGINEER



IN THIS ISSUE

• FLASH STEAMERS • CUTTING CYLINDER LATCH KEYS AN ELECTROMAGNETIC CLUTCH • QUERIES AND REPLIES • LAYOUT OF STEPHENSON LINK-MOTION • BALANCING A LATHE COUNTERSHAFT PULLEY • A DIVIDING DEVICE

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THE MODEL ENGINEER

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CONTENTS

Our Cover Picture

This photograph came to us from far-away Australia; it depicts a 10¼-in. gauge reproduction of a Gresley L.N.E.R. Pacific locomotive belonging to Dr. Herbert Busch. The engine is stated to be coke-fired, which, to us, is not usual; but probably our Australian friends find that they obtain better results from coke than from the local coal.

Engine and tender are 12 ft. 6½ in. long and operate on a 440-ft. track. We are interested to note that English locomotives have their "fans" in Australia, judging from this and other photographs that we have seen. Incidentally, the hobby of building miniature locomotives and railways seems to be steadily gaining favour "down under," and we are beginning to wonder if the time is not approaching when annual "test matches" could be arranged to take place between Australian and English miniature locomotives!

SMOKE RINGS

Time is Nearly Up

THE "M.E." Exhibition is now only a fortnight away and the finishing touches are being made to models all over the country, in order that they may be in tip-top condition for the show. Our friends of the trade, too, are making sure that their display will be at least as comprehensive as ever.

Our usual articles under the heading of "What to See at the 'M.E.' Exhibition" will give our readers a brief preview of what is to be found there; practically every phase of our hobby is represented by new work that has been produced in less than twelve months since the previous exhibition.

Except on the opening day, when the public will be admitted at 2 p.m., the hours are 11 a.m. to 9 p.m. daily, omitting the Sunday.

New Uses for Plastics

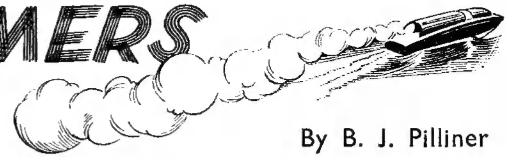
AT THE British Plastics Exhibition held recently at Olympia, many improvements were in evidence not only in the composition of plastic materials, but also in manufacturing technique and scope of applications. The uses of plastics have extended far beyond those which were obvious, or originally visualised; they are rapidly finding new, and in many cases, unexpected applications. In engineering foundry work, for instance, synthetic resins have not only become serious rivals to the gums and other binders used for strengthening cores, but they can now be used for external moulds as well, and the new "shell moulding" technique enables both accuracy and finish to be considerably improved in all kinds of metal castings. Another notable modern development is the method of building up sheets of any thickness and three-dimensional contour, by the use of these resins as a binder for laminated fabrics having either a mineral or vegetable base. The range of plastic raw materials, for production by pressure or injection moulding, casting, machining, or fabrication, or as ingredients in paints or varnishes, has increased and machinery for production has vastly improved. Many of the difficulties or disadvantages in plastics production have been or are being surmounted, and they have long since ceased to be regarded as mere substitutes for wood, metal, leather, etc. Small articles of practically any kind are specially suited for production in plastic materials, and there is considerable scope for their use in model engineering, where their ease of working, strength, or durability, and elimination of the need for surface finishes, are valuable assets. Unfortunately, however, a present obstacle to their extensive use in the amateur workshop is the difficulty of obtaining retail supplies.

Our Draughtsmen Bow

IT OFTEN happens that contributors send us articles which deal in masterly manner with particular subjects but accompanied by mere sketches, sometimes in pen-and-ink though usually in pencil. In cases of this kind, the sketches are handed over to our draughtsmen who set to work to prepare drawings that will give the best possible results when reproduced in the pages of THE MODEL ENGINEER.

Proofs of the articles and illustrations are then sent to the contributors, and almost invariably a letter comes back to thank us for "having made such an excellent job of my poor sketches," or words to that effect. This sort of thing, naturally, always gives pleasure to our draughtsmen; but after all, it is their job, since they know what sort of drawing will give the best reproduction. There is not the least necessity for any potential contributor to think that he cannot write or draw well enough to stand the test of publication; if he makes himself clear enough, in both cases, we can do the rest.

FLASH STEAMERS

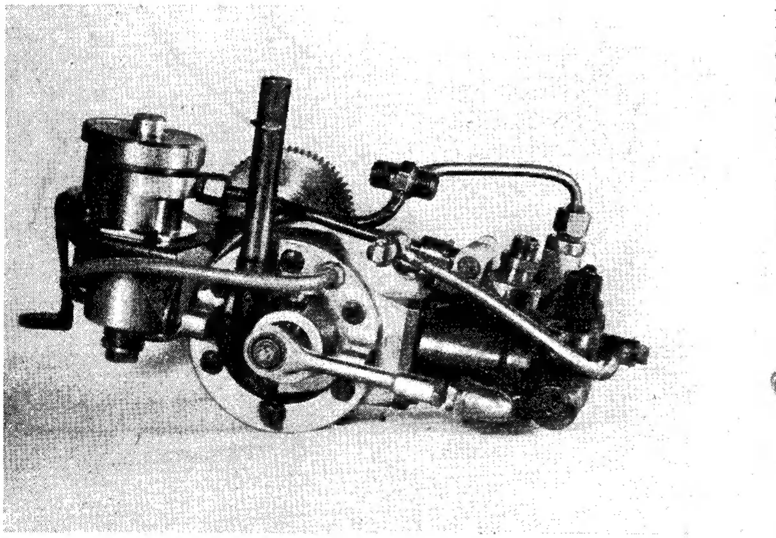


● A CHRONICLE OF EXPERIMENTS, TRIALS AND TRIBULATIONS

By B. J. Pilliner

THE main changes in engine design have been from twin-cylinder to single-cylinder of reduced capacity, and from poppet valve uniflow to piston valve. Due to the high pressure at which it is possible to run a flash boiler without risk, it is possible to get a great deal of power from a single-cylinder of modest dimensions, and there is no reason to suppose that the output of the latest engine cannot be increased by using steam at higher pressure, if the lamp and boiler can be persuaded to increase the steam supply.

Valve Gear. The first choice of poppet valve with uniflow exhaust was made for the following reasons: (a) No risk of seizure ; (b) Negligible steam leakage ; (c) Pressure lubrication to steam supply unnecessary.

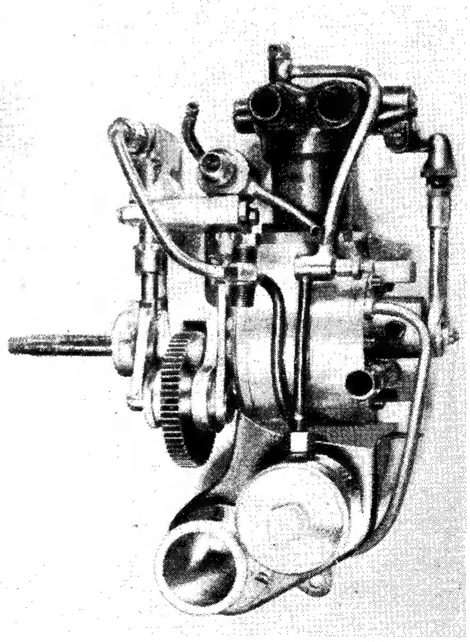


A view of " Frolic's " engine from the forward end

Lubrication of the poppet engines was by splash from the crankcase, and at the low power obtained was effective, but I doubt whether upper cylinder lubrication would be satisfactory by this means alone, with the power and steam temperature of later boilers. (All engines have used ball-races extensively, making pressure lubrication to bearings unnecessary.)

The poppet valve was abandoned, owing to suspected valve float as revs were increased. At first sight it would seem that this valve should be at least as satisfactory as its counterpart in an equivalent petrol engine at the same r.p.m. It has the advantages that it may be smaller, lighter, and have less lift than for a petrol engine, but these points are more than outweighed by the fact that it has to open and close in much less time, with consequently higher acceleration and greater mechanical stresses. If cut-offs earlier than half stroke are required it has only one third, or less of the time to open and close, compared to a four-stroke petrol engine valve at the same r.p.m. For example, assuming that valves in both engines are identical, and are operated by similar gear, then reducing the opening period to one-third increases the acceleration by $3^2 = 9$. For the sake of argument the valve lift may also be reduced to one-third, reducing the valve acceleration in direct proportion, but the maximum valve acceleration is then still three times greater for the steam engine. Opening and closing of the poppet valve are also slow compared to the piston valve, an undesirable characteristic which could be overcome by using a masked poppet valve, which would also lower the acceleration. However, one is then halfway to a piston valve, as a masked valve introduces the risk of leakage when the valve is off its seat but not open.

When the poppet valve was given up, a great deal of thought was given to the choice of a gear to replace it. It was then realised that the valve gear is the heart of the high-speed flash-steam engine, and that no known gear is ideal in all respects. Once the valve gear had been decided, the rest of the engine was comparatively simple, as established petrol engine practice was followed fairly closely. The piston valve was chosen as being the only remaining suitable type, and it was decided to discard the uniflow principle, as it is difficult, if not



Right—Plan view of engine

Continued from page 126, July 30, 1953.

impossible, to run a flash steamer at predetermined conditions, and a characteristic of the uniflow is that it should be designed and run at a particular pressure and temperature.

The conventional type of single piston-valve gear, driven by an eccentric on the crankshaft, is rather unsuitable, owing to the limitations which it imposes on the timing of valve events, and the small steam opening relative to the stroke of the valve when timed for early cut-off. Some people have overcome the first of these difficulties by using two valves for steam and exhaust, and a suitable valve diameter and stroke no doubt overcomes the latter. However, I wished to avoid two valves if possible, owing to the increased steam, leakage which it implied. I therefore concentrated on devising a driving gear for a single valve which would give an acceptable timing and port opening. First ideas were in the nature of moving the valve by complementary cams, a method which is, of course, by no means new, but this scheme was discarded owing to the possibility of wear on cams and followers, with the consequent difficulties of keeping the mechanism free from backlash, and hammer action.

The valve gear eventually decided upon is still in use on the engine in its present form. A single piston valve of $\frac{3}{8}$ in. diameter, mounted horizontally on top of the cylinder, is driven by an eccentric or crank via the usual valve connecting-rod and bell crank. The eccentric is mounted on a shaft which takes its motion from the crankpin of the overhung crankshaft by means of a slotted arm, fixed to the shaft, engaging a square die block pivoted on the crankpin. A varying angular speed is given to the eccentric, by placing the shaft on which it is mounted, off centre to the crankshaft. This mechanism is actually a variation of the well-known quick-return motion, common on shapers, with the constructional difference that the lever, or arm, pivot point is inside instead of outside the crank circle, resulting in the lever being given a rotary instead of an oscillating motion. The angle of lead of the eccentric, relative to the crank, varies as the crank rotates enabling the timing of valve events and the amount of inlet valve opening to be improved. The diagrams compare the valve timing of this method with the orthodox method, and are approximate in so far as they are not corrected for angularity of bell crank and connecting-rods.

The main disadvantage of this valve drive is the considerably increased mechanical loads, due to the increased accelerations of the various parts. Since first fitting this valve gear, there have been several mechanical failures, and all parts have been improved and replaced at least once, some of them several times. The valve travel, which was originally $\frac{3}{8}$ in., was later reduced to $\frac{1}{4}$ in., and the mechanism in its present form is reasonably safe up to approximately 9,000 r.p.m., but prone to failure due to over-revving when the propeller leaves the water on a "flip." I may add that Mr. F. Jutton, finds this type of valve gear quite satisfactory on his engine. In this case, the valve is arranged parallel with the cylinder, eliminating the bellcrank, and shortening the valve connecting rod. It would be interesting to know if the efficiency of the side steam entry to the cylinder is appreciably lower than the central entry of my engine, which on this account has to move the greater weight of valve gear. Bench testing would be the only means of finding the answer to this query.

Engine Details

The piston valve engine fitted to *Frisky*, although rebuilt to a large extent for *Frolic*, has been altered and developed in detail only. The original 3 per cent. nickel cylinder had auxiliary exhaust ports opened by the piston at the bottom of the stroke. This caused trouble in the event of piston ring breakage, and as covering them up made no noticeable difference in performance they were omitted when making a new cylinder for *Frolic*. Piston clearance of 0.007 in. at the top and 0.005 in. at the bottom is necessary to avoid signs of tightening up, and when using auxiliary exhaust ports, the piston ring was positioned as near the top as possible, and the piston chamfered almost to the ring groove. In this position the ring gets overheated and loses its spring after only one or two runs. *Frolic's* cylinder is of cast iron and the piston surface has "bedded" down well, which was not the case with the steel cylinder. The single $\frac{1}{16}$ in. wide piston ring (Wellworthy), is fitted just above the gudgeon pin, in which position it is not affected by heat, and is replaced only when the gap has increased appreciably.

Ball-races. The standard extra light journal series are used. Crankshaft $\frac{5}{8}$ in. and $\frac{3}{8}$ in., big end $\frac{3}{8}$ in., valve drive shaft and pump drive shaft $\frac{1}{4}$ in., valve crank $\frac{3}{16}$ in. The last is not listed in the above series but is apparently a standard race. The larger crankshaft race was originally $\frac{1}{2}$ in., changed to $\frac{5}{8}$ in. when the engine was rebuilt for *Frolic*, as it now takes the propeller thrust. A few comments on the most suitable types of races may be of interest. A "crowded" race (full of balls with no cage), is unsuitable for thrust owing to the filling slots. Caged journal races can take a certain amount of thrust, according to the makers' recommendations. The type of cage which is made of thin steel and curled over between the balls from one side seems to be completely unsatisfactory for high speeds. I have had little trouble with the more usual two-piece brass cage, which covers the balls on both sides, although I would not now trust it on the big end, for which a "crowded" race is perfectly safe.

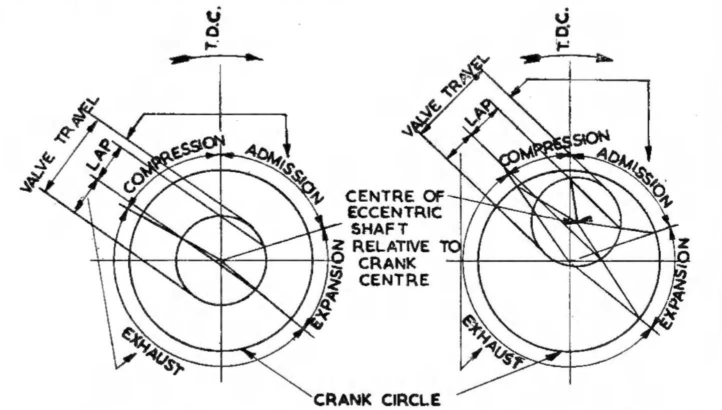


DIAGRAM FOR ENGINE WITH ECCENTRIC ON CRANKSHAFT ADMISSION, CUT OFF, AND RELEASE POINTS AS IN 'FROLIC' DIAGRAM

VALVE DIAGRAM 'FROLIC' INCREASED ADMISSION OPENING, REDUCED COMPRESSION PERIOD

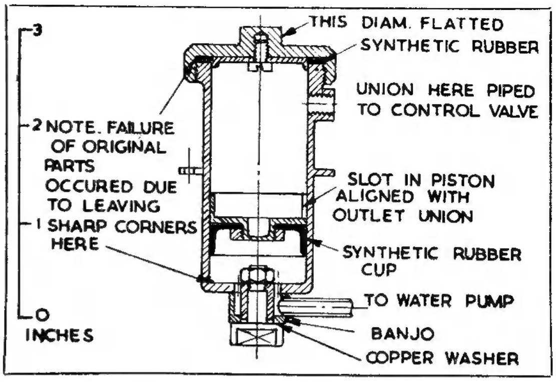
A disintegrated big end makes a terrible mess of about half the engine, as I found when using one of the above-mentioned steel cage type. The life of the races is probably governed as much by deterioration due to moisture as by wear, the big end having the shortest life, and lasting about a season. As much moisture as possible is removed from the engine after running, and it is swamped in oil. At the same time the plant is checked over generally, paying particular attention to any items which have currently been giving trouble. Although the necessity for this does not say much for the general reliability, trouble is often detected at an early stage and further damage avoided.

Materials.—Alloy steel, heat treated to about 60 tons is used for crankshaft, valve drive arm, valve crank, valve bell crank and gudgeon pin. The cylinder and valve are "Brico" centrifugally cast iron, the latter having a steel insert at the driving end. Crankcase parts, piston connecting rod, valve connecting rod and pump mounting are dural.

Engine mounting. The method of mounting the engine and oil tanks in the hull of *Frolic* is very rigid, and fractured oil and water pipes, prevalent with its predecessors, are now a thing of the past. The rear end of the crankcase fits closely in a ring bolted to the transom, and a dural mounting ring bolted to the front of the crankcase has a 2-B.A. hole each side for socket-head screws securing it to the engine plates. Each engine plate is attached to the wood bearer, by four 6-B.A. bolts and backing plates,

Engine plates have fractured more than once during a rough run with the propeller leaving the water, and have been changed from 18 g. M.S. to 16 g. stainless steel. The mounting bracket for the oil tanks, originally 18 g. aluminium alloy, which fractured with vibration, has been changed to 20 g. M.S. It is bolted to the crankcase and engine plate and has a bolting point for a 2-B.A. screw on the nearside of the hull.

Lubrication. Two oil containers are now used, one feeding motor oil by centrifugal force, after the boat has started, to the crankcase, and the other containing superheat steam oil, which is fed to the incoming steam on the cylinder head. The pressure system for the steam oil was adopted with the object of avoiding the complication of adding an oil pump when changing to piston valve. The container is filled with steam oil and has a free piston which is then at the bottom. Water from the output of the water pump is fed to the bottom and forces out oil at the top, through a needle control valve, to the cylinder-head. The free piston acts as a separator for the oil and water, and is pushed down before topping up. When making runs at close intervals, the quantity of oil fed tends to increase, due to the container picking up heat from the rest of the plant, which lowers the oils viscosity and allows the needle valve to pass more oil. For this reason the valve setting is usually reduced after the first run, but consumption is not critical, and there appears to be no difference in performance between the extremes of using one-quarter or three-quarters of the oil in the container. Some oil fed in with the steam is essential, and once, when the oil pipe to the engine fractured, the valve seized in a few seconds.



"*Frolic*"—*pressure lubricator. Materials: Container, cap and piston—dural. Banjo—M.S.*

It should perhaps be explained that this oiling system has nothing in common with what is generally known as a displacement lubricator, but works on the principle that as long as there is a flow in the boiler there is a difference in pressure between the two ends, the water pressure being the greater. It is the difference in pressure which is available to feed the oil.

Mechanical Water Pump

As previously mentioned, this is driven by a reduction of 2.8 : 1 from the engine, and is generally conventional in design, with stroke adjustment by means of a choice of several holes in the crank disc. This adjustment is from $\frac{3}{8}$ in. to $\frac{5}{8}$ in.; the bore is 9/32 in., and the stroke at present in use is $\frac{1}{2}$ in. Valve boxes are bronze, and the $\frac{3}{16}$ in. diameter stainless steel balls are restricted to 0.015 in. lift. The body is dural, and the ram stainless steel. The pump has given very little trouble, and is assumed to be reasonably efficient, as the response to small changes in stroke is noticeable on the running of the boat.

For the benefit of those not familiar with the details of flash steamers, I will explain that the water filter is normally positioned in a small tank, which fills up when the boat is running. The contents of this tank feeds the pump when the boat bounces on rough water. This is a necessary device, as admission of air to the pump when working at a high pressure is likely to cause a complete fade-out of the plant: the pump cannot lose the air, which is alternatively compressed into the clearance volume and then expanded, and returning the scoop to the water cannot help. The main points of the water tank are that the inlet from the scoop is at the top, the outlet to the pump at the bottom, and that a pipe is taken from the top to a position outside the hull, which is submerged when starting, and above the water when running. This pipe acts as a vent for any air which finds its way into the tank, and as an overflow for surplus water picked up when the tank is full. The action of centrifugal force was taken into account when positioning the various tank openings, as on a 100 yd. course, this amounts to more than 5 g. at 60 m.p.h., and the resultant between this and gravity is approaching the horizontal. In these circumstances the words "top" and "bottom" used above can be regarded as nearside and offside.

The scoop shown in the drawing, which consists merely of a vertical pipe cut off at an angle, and designed some years ago to avoid blockage from leaves, etc., has recently been replaced by a forward-facing scoop, positioned at the rear edge of one of the front planes. The intention of this change was to lessen resistance, which at higher speeds must be considerable with the former arrangement. The water tank of *Frolic* is primed with a syringe, via the overflow, before starting, a precaution found necessary to avoid the air locking trouble mentioned above, which sometimes occurred on starting the boat.

(*To be concluded*)

L.B.S.C.'s "Britannia" in 3½ in. Gauge

• PLUMBING JOBS

NOW is the time to call in the plumber! The position of the injector, down by the ashpan on the left-hand side, between the trailing coupled and pony wheels, is shown on the general arrangement drawing, so no detail illustration is needed. The water pipe from the injector water valve, shown in last instalment but one, goes direct to the injector, and is connected to it by an inverted swan-neck with the usual union nut and cone. I've explained how to make these so many times, that readers should be able to do it with their eyes shut! The 5/32-in. steam-pipe is connected to the injector steam-valve at the side of the firebox wrapper, passes down under the cab deck, across backhead, and turns around to the injector, as shown in the instalment mentioned above. It is coupled to the steam end by a union nut and a flat collar, as usual.

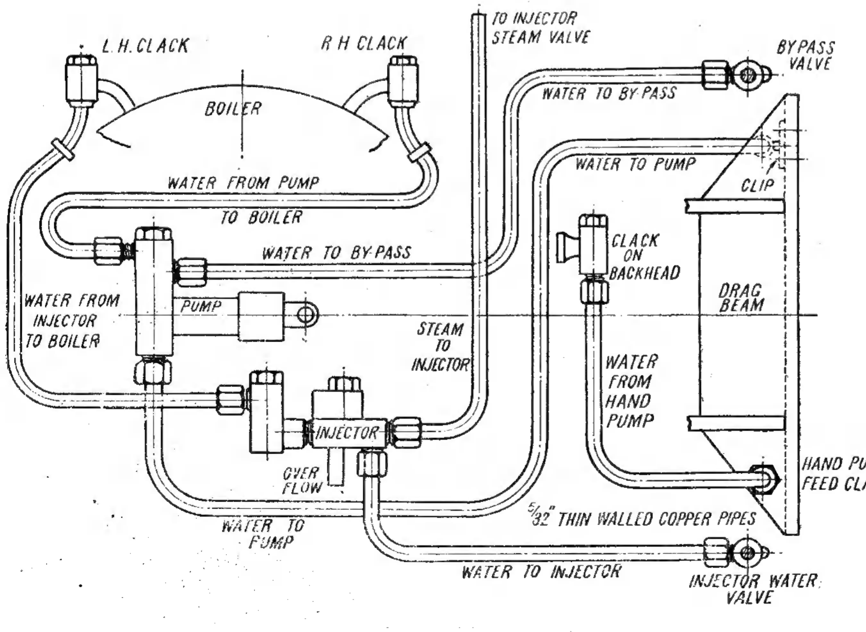
The delivery pipe goes from the union on the check valve, to the top-feed clackbox on the left-hand side. Measure with a bit of thick lead fuse wire, or soft copper wire, from the union on the clackbox to the flange, or union, whichever was fitted, on the top clack. This pipe takes an upward slant from the injector to the running-board, passes through same, and runs along the top of it until almost level with the clack, then turns upward in an easy bend, hugs the boiler-barrel and joins the clackbox. Straighten out the wire, cut a piece of pipe to same length, fit the unions, or union and flange as the case may be, drill a hole in running-board big enough to let the union nut pass, insert pipe, bend to contour, couple up both ends, and Bob's your uncle. If a flange joint is used at the top clack, put a 1/64 in. Hallite or oiled paper joint between the flanges, and don't forget to punch a hole in the middle!

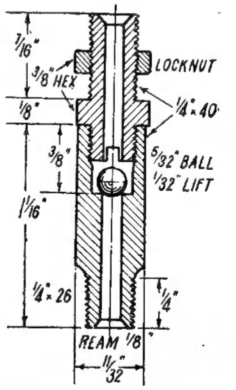
Feed and Bypass Pipes

As already explained, the bypass valve is the same as the injector water-valve, only on the opposite side; so the only job is to run a 5/32-in. pipe from the bypass union on pump, under right-hand running-board, down side of firebox, and under side flange of ashpan, to the union on the valve. The pump feed pipe is connected to the union under the pump valve-box; and to save unnecessary length of pipe, it may be run along inside frame, to boiler throatplate, coming out over the rear frame stay, and then keeping the bypass pipe company, right back to the valve. Bend it down at this point, so that the end of the pipe will be parallel with the piece of pipe below the bypass valve; the two connecting hoses, or "feed-bags," will then be in line when the engine and tender are coupled up. These vertical connections allow far more freedom for both side and fore-and-aft movement, than if the pipes were horizontal; this is a great advantage when the engine is lengthy and there is plenty of overhang on curves. A clip will be needed to hold the pipe in position (see detail sketch), but nobody should need detailed instructions on how to bend a $\frac{3}{16}$ in. strip of brass into a clip, and fix it to the drag-beam with a couple of nutted screws.

Pipes and Clacks for Hand-pump

We shall carry the usual "boiler insurance policy" in the tender, and this will need a delivery clack,

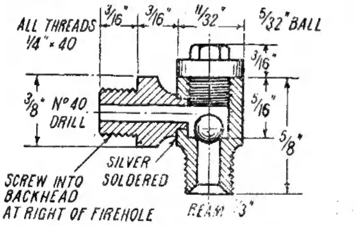




Section of hand-pump clack

Left — Diagram of pipe connections

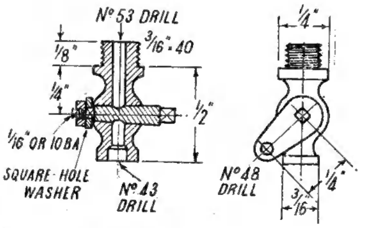
with pipe and union for connecting up to the feed-pipe. In districts where the water is chalky or otherwise impure, clacks on the backhead often start to leak, due to grit and scale on the ball seating; and the union not only gets very hot, as you find if you try to disconnect it with your fingers, but hot water finds its way back to the tender tank, and warms up the feedwater sufficiently to prevent the injector from working. I am therefore specifying another clack which is combined with the union for the connecting pipe. The backhead clack is just one of my standard type, made to the sizes given in the illustration. These have been described so many times that full repetition is needless. The inside is made exactly the same as the top feed clacks, and the stem is screwed for a $\frac{1}{4}$-in. × 40 union. A fitting



Section of backhead

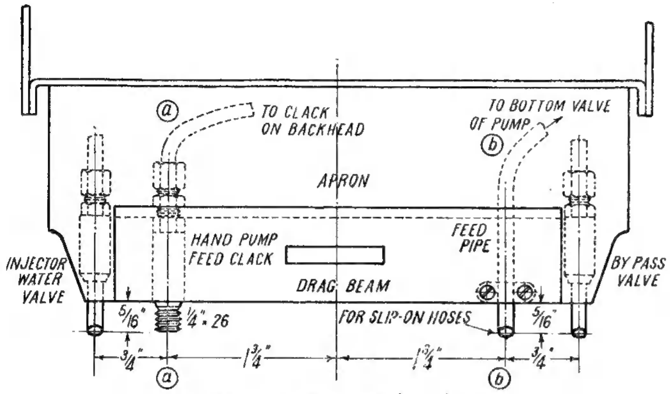
made the same as that at the top of the water-gauge, is silver-soldered into the side. Drill a 7/32 in. hole in the backhead, to the right of firehole, and about $\frac{3}{8}$ in. below the whistle-valve spindle, midway between firehole and edge of backhead. Tap it $\frac{1}{4}$ in. × 40, and screw the clack in with a smear of plumbers' jointing on the threads.

To make the bottom clack, chuck a piece of $\frac{3}{8}$-in. round brass rod in the three-jaw. Turn $\frac{1}{4}$ in. length to $\frac{1}{4}$ in. diameter, and screw $\frac{1}{4}$ in. × 26. Centre deeply, but don't drill; part off at $1\frac{1}{16}$ in. full from the end. Reverse in chuck, centre, drill through No. 34, open out and bottom to $\frac{3}{8}$ in. depth with 7/32-in. drill and D-bit, and tap $\frac{1}{4}$ in. × 40. Ream the remains of drill hole with $\frac{1}{8}$ in. parallel reamer. Seat a 5/32-in. ball on the hole, and gauge depth from ball to top. Chuck a bit of $\frac{3}{8}$-in. hexagon rod in three-jaw, face the end, and turn down to $\frac{1}{4}$ in. diameter for a distance equal to 1/32 in. less than indicated by the depth gauge. Part off at $\frac{9}{16}$ in. from the shoulder, and reverse in chuck, holding either by the hexagon or in a tapped bush. Turn down $\frac{7}{16}$ in. length to $\frac{1}{4}$ in. diameter, and screw $\frac{1}{4}$ in. × 40. Centre deeply. and drill right through with No. 34 drill; cross-nick the bottom with

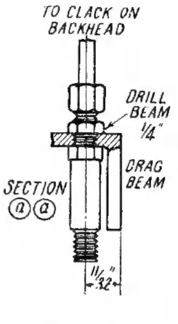


Cylinder drain cocks

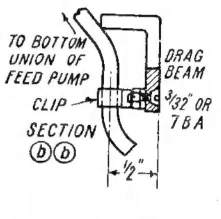
a thin file. Assemble as shown, putting a smear of plumbers' jointing on the threads; then make a locknut from the $\frac{3}{8}$-in. hexagon rod. Drill a $\frac{1}{4}$ in. clearing hole in the top of drag beam, approximately $\frac{9}{16}$ in. from the left-hand end, in the angle as shown in plan. Put the long stem of the clack through from underneath, and secure it with the locknut. Connect the top of the clack above the beam, to the stem of the clack on backhead, by a piece of 5/32-in. pipe with union nuts and cones at each end, as shown in the diagram, keeping the vertical part of the pipe as close to the backhead as possible.



Pipe connections at drag beam



How to erect hand-pump clack



How to erect feed-pump connection

The vertical union will be found easy to couple up, and it also gives more flexibility to the tender pipe on an engine of the length of *Britannia*; but if anybody prefers a horizontal union, don't form the union screw at the bottom of the clackbox, but leave it full size and drill it "blind." Drill a $\frac{3}{16}$-in. hole in the side, close to the bottom, and in it, silver-solder a $\frac{1}{4}$-in. × 26 union screw. Erect with this screw pointing to the rear.

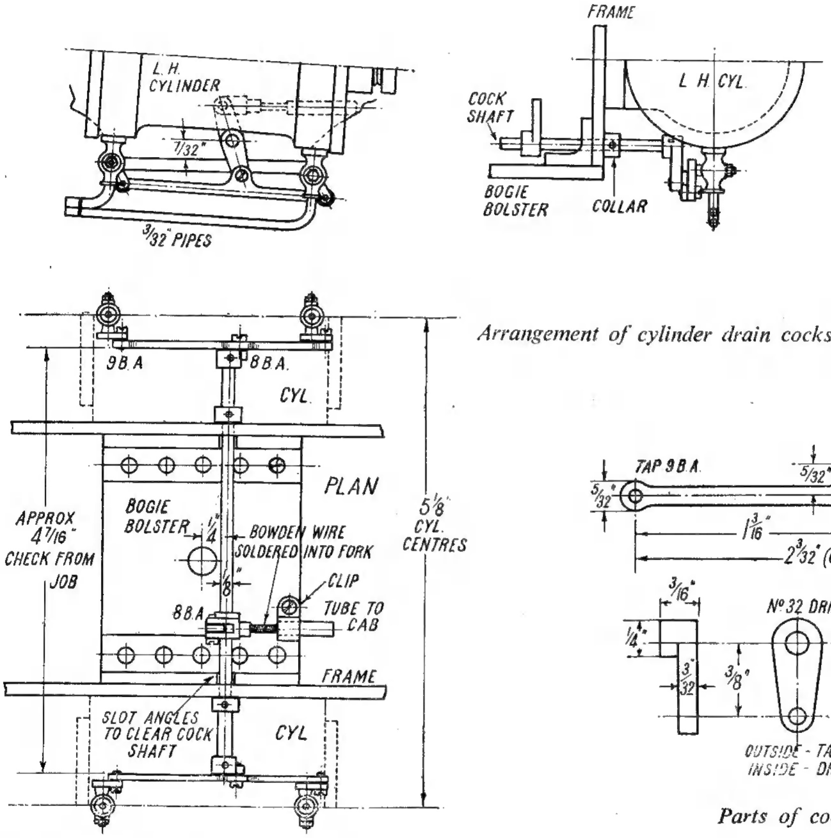
Cylinder Drain Cocks

We shall, after all, have to fall back on the arrangement of cylinder drain cocks specified for *Doris* and *Pamela*. It is hopeless to try to make the steam-operated cocks as a working proposition, due to the length of small-diameter pipe from cab to cylinders, in which the steam would condense. The pipe would keep hot all right *if steam were passing through it* all the time; but with the steam "static," condensation would take place, and put the cocks out of action. If a short pipe from the smokebox end were connected to steam-operated cocks, a valve would be required at the front end to operate them, and this would need a rod or wire from the footplate, in any case; so the steam gear might as well be cut out, and the cocks operated direct from the wire, thus saving unnecessary complication.

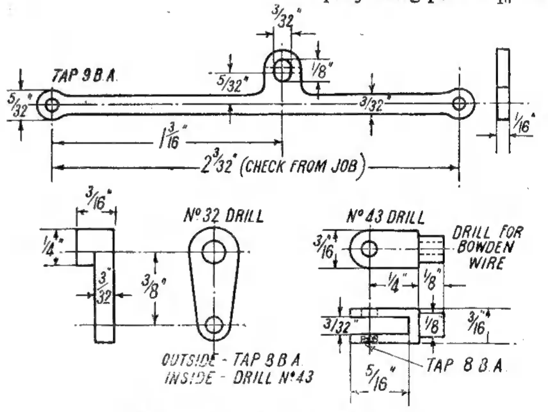
The process of cock-making has already been fully dealt with in previous notes; in the *Doris* serial, for example, but a condensed version of the way I usually do the job, may

help new readers. The bodies are turned from $\frac{1}{4}$ in. round bronze or gunmetal rod; if I specified hexagon, the fat would be in the fire at once, as my old granny used to say, because big engines don't have hexagon cock bodies. Chuck the rod in the three-jaw, turn down $\frac{3}{16}$ in. length to $\frac{3}{16}$ in. diameter, and screw $\frac{3}{16}$ in. × 40; then face off the end until the screwed part is a full $\frac{1}{8}$ in. long. This ensures a full thread right up to the end, which is advisable as the holes in the cylinder flanges are shallow. Part off at $\frac{1}{2}$ in. from the shoulder, and ditto-repeato six times, as you might as well have a couple of spares while on the job !

The easiest way to cross-drill the bodies, is to use the method I described for drilling handrail knobs. Drill a $\frac{1}{4}$-in. hole in the middle of a piece of $\frac{1}{2}$-in. square rod, and at $\frac{1}{4}$ in. from the end, drill a No. 43 hole exactly across it. Push each blank into this, with the shoulder level with the end, then poke the No. 43 drill down the guide hole, and carry on right through the blank.



Arrangement of cylinder drain cocks



Parts of cock operating gear

Chuck each blank in a tapped bush held in three-jaw, and turn to the outline shown. Centre, drill right through with No. 53 drill, and counterbore No. 43.

Cock Plugs

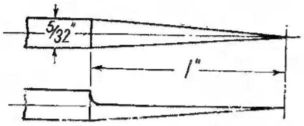
First make a taper reamer as shown, by the same process as described for injector reamers; but after turning the taper, *don't shift the top-slide, nor alter the tool.* Put a tapwrench on the shank, and ream the cross-hole in each cock body, but leave a bare $\frac{1}{16}$ in. of the end of the hole, parallel; this allows for future grinding, if the plugs should develop leakage. Now chuck a piece of $\frac{1}{8}$-in. bronze or gun-metal rod, and with the top-slide as set for turning the reamer, turn a taper on the end, long enough to allow a full $\frac{1}{8}$ in. to project when a cock body is tried on it. If your top-slide has a four-tool turret, slew the parting-tool around, and part off at $\frac{1}{2}$ in. from the cock body. If the slide only carries a single tool, don't shift it, but cut off the plug blank with a fine hacksaw. Ditto-repeato six times, then set the slide to turn parallel again. Chuck each blank in the three-jaw, turn 3/32 in. of the small end to $\frac{1}{16}$ in. diameter, and screw $\frac{1}{16}$ in. or 10 B.A. File the next $\frac{1}{16}$ in. to a square. Push back the blank in the chuck until there is only $\frac{1}{16}$ in. projecting between the chuck jaws and the shoulder; use a " safe-edge " flat file (one with no teeth on one edge) and keep the safe edge in contact with the jaws while filing. Three or four light strokes with No. 1 chuck jaw set in the 3, 6, 9, and 12 o'clock position, will do the trick nicely.

To hold the plugs for squaring the other end, make a taper bush. Chuck a bit of $\frac{1}{4}$-in. round rod, $\frac{1}{4}$ in. long, in the three-jaw, centre, drill through No. 43, and ream it same as the cock body, so that when a plug blank is pushed in, it will project through as far as the end of the squared part next the screw. Push each blank tightly into the taper bush, part off, or face off, until the projecting part is $\frac{3}{16}$ in long, then file a square $\frac{1}{16}$ in. long on the end, using the chuck jaw as a divider, as mentioned above.

Assembly

The handles or levers are filed up from $\frac{1}{16}$ in. × $\frac{1}{4}$ in. steel strip, to the shape shown. Drill the larger end No. 43, and file the hole square, by aid of a small square watch-makers' file, so that the lever fits tightly on the squared end of the cock-plug. They can be prevented coming off by aid of a $\frac{1}{16}$ in. or 10-B.A. screw and washer; or a touch of solder is quite satisfactory. I usually burr over the edge of the

square very slightly, but it needs care to avoid damage to the plug. If screws are desired, drill and tap the hole in the end of the plug, while it is still in the taper bush after the filing process. Fit a plug in each cock-body, securing with a commercial nut and a square-holed washer. To make washers, just chuck a piece of 5/32-in. round brass



Cock plug reamer

rod, centre, drill No. 50, and part off $\frac{1}{16}$-in. slices. Hold each in pliers by the edge, and a slight application of the watchmakers' file does the trick.

To drill the plugs, set two of the levers pointing left as shown, and two pointing to the right; then poke the No. 53 drill down the hole in the cock-body, and carry on right through the plugs. When in place on the engine, all four cocks must be open or shut, when all four levers point in the same direction.

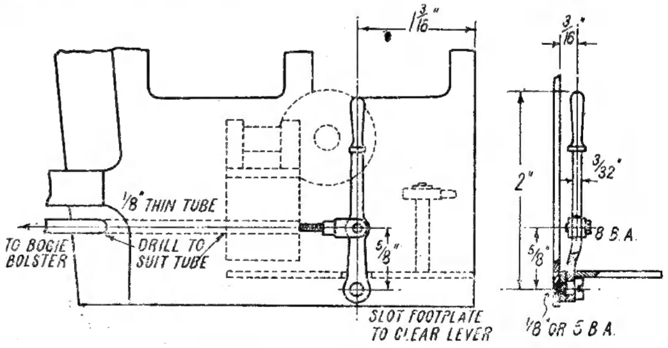
If the plugs are anointed with a very slight smear of the finest grade of valve-grinding paste, or a scraping off your oilstone, a few twiddles back and forth, pressing very lightly, will grind them in all right. *Don't use metal polish*, as the surfaces should not be smooth, but matt. They will then retain an oil film, and the cocks will work easily, yet remain steam-tight. On the L.B. & S.C.Ry. we used to grind in the gauge cocks with powdered bath-brick, silver sand, or similar abrasive, and water, carefully washing off all grit when through. They were given a dose of tallow or cylinder oil, and graphite, when reassembling; they operated with little effort, and didn't leak. Be careful to wash off all traces of the grinding compound before fitting the cock plugs "for keeps." A spot of cylinder oil, plus a taste of graphite (a scraping off a soft lead pencil will supply this) will ensure easy working. Put a smear of plumbers' jointing on the threads of the body, and screw home into the tapped holes in the cylinder flanges, being careful to put them in so that they are all open when the handles point forward, and shut when the handles point backward.

Operating Gear and Erection

The rods connecting the levers are shown in the drawing, and need no detailed explanation; file them up from 16-gauge steel. The slotted hole is needed, owing to the different radii of the end and middle levers. Attach by 9 B.A. screws to the cock levers at each side, and measure between the inner sides, to get the exact length of the cock shaft, which is made from $\frac{1}{8}$-in. round steel. The levers, shown in detail drawing, can be filed or milled up from $\frac{3}{16}$-in. $\times \frac{1}{4}$-in. steel. In each side frame, under the cylinder, drill a No. 30 hole through frame and bolster angle, 7/32 in. from bottom of frame, and $\frac{1}{4}$ in. behind the centre line of bolster; this should clear the screws. Temporarily remove bolster, and file slots from the top of the angles, to meet the holes; this will allow the bolster to be replaced after the cock shaft has been erected. Put the shaft through holes in frame, putting one lever between frames, and one on each end; a collar (slice of $\frac{1}{4}$-in. rod with a No. 31 hole through it) goes between lever and frame. Adjust the levers so that they are parallel, and just clear of the rod connecting the cock handles, and pin them to the shaft. Set collars against frame, and pin them too. The whole doings is shown in plan drawing. Personally, I use No. 56 drill and bits of steel blanket pins; but ordinary silver-steel may be used if desired. The inner lever is set exactly opposite the outer ones (it is shown dotted in the elevation drawing) and just clear of the bolster fixing angle. Pin it, same as the others, but leave the bolster off until erection of cock gear is finished.

filed in the cab deck or footplate, to clear the lever. Connection is made between cab and cock levers by a Bowden wire running through a thin $\frac{1}{8}$-in. tube, the wire being furnished with a fork at each end; the wire is soldered into the boss of fork. The complete outfit is shown in the illustrations, and needs little explaining. The length of the tube is easily obtained from the actual job; it starts from the bolster, takes an easy slant upwards to top of frame between the driving and trailing coupled wheels, swerves out by an easy curve, and makes a beeline for the cab, into which it passes through a hole in the front plate, and finishes up at the reverser bracket.

The Bowden wire is the same as used for brakes on pedal cycles, and can be obtained from any cycle shop. Solder a fork on one end, and push it through the tube from the bolster end, connecting the fork to the cock lever as shown. You can then see exactly where to cut it, to attach the fork at the cab end. Solder on the other fork, and attach to the cab lever by a setscrew as shown. When the cock levers are in mid-position, the cab lever should be vertical. Replace the bolster, and make a clip to screw to it, from a strip of brass, to support the tube as shown, attaching the clip to bolster by a 3/32-in. screw. The tube should need no support between bolster and cab; but if anybody



How to erect cylinder cock lever

The lever in the cab can be made from $\frac{1}{4}$-in. square steel, turning the grip in the four-jaw, and milling or filing the lower part to shape shown; or the flat part can be made from 3/32-in. $\times$ $\frac{1}{4}$-in. strip, and the grip turned from $\frac{3}{16}$-in. round rod, and brazed on—my pet way. It is attached to the inside of the cab side sheet by a shouldered screw at the bottom, which can be turned from $\frac{5}{16}$-in. round rod. A slot is fancies to clip the tube at any place along the route, there is, of course, not the slightest objection. The Bowden casing (which is like a coiled spring) on the cycle brakes, is usually left free between the ends. If the wire is greased before inserting it into the brass tube, the cocks should operate quite freely, but the driver's handle will "stay put" in either on or off position, and no stops will be required.

AN ELECTROMAGNETIC CLUTCH

By R. F. Stock

THE complete bobbin was now pushed into its annular recess in the clutch body, the connecting wires being carefully entered into the drilled holes. Before the bobbin was right home, a quantity of shellac was poured into the recess and this was sufficient completely to fill all spaces around the coil. The object is, of course, to give the effect of embedding the wires in a solid block.

The slip-ring assembly was then bolted to the rear face, the connections being carefully fed through their holes. The ends of the wires were then cut to length, bared and threaded through the slits in the plastic; they were pressed into the slots cut in the sides of the brass rings and soft-soldered. The final move was partly to fill the hollow slip-ring carrier with a thermo-plastic to protect the connecting wires and also to lock the retaining-bolts. This plastic filling is shown cross-hatched in Fig. 5.

The weight of the completed clutch was driving member $13\frac{1}{4}$ oz., driven member $5\frac{3}{4}$ oz., total 19 oz.

Concluded from page 110, *July* 23, 1953.

Testing

When completed, the driving member of the clutch was locked to a $\frac{1}{4}$-in. silver-steel rod held firmly in the bench vice. The driven member was loose on the shaft, and soldered to it was a brass plate screwed to a balsa beam 18 in. long. A screw eye at the tip enabled a spring-balance to be attached, to apply a force tangential to the end of the beam.

Two carbon brushes were supported in a temporary holder and spring-loaded against the slip-rings; they were connected to a source of variable voltage (d.c.) and a high resistance voltmeter was connected across them.

For the first test the faces of the clutch were smeared with engine oil, and torque noted against various voltage readings. In all cases three readings were obtained at each value of applied volts—the force required to overcome static friction in the clutch, the force required to maintain a slow smooth rotation of the beam and the latter reading after the voltage was cut off.

This last force was found to be nearly constant and, reduced to torque, had a value of $2\frac{1}{2}$ lb./in.

The static friction was not easy to obtain

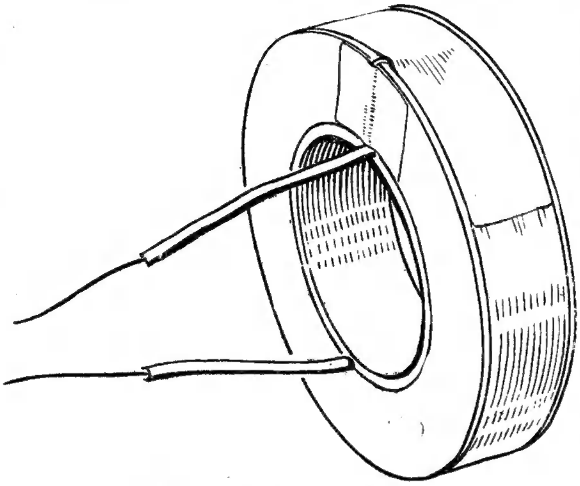


Fig. 7. The completed coil

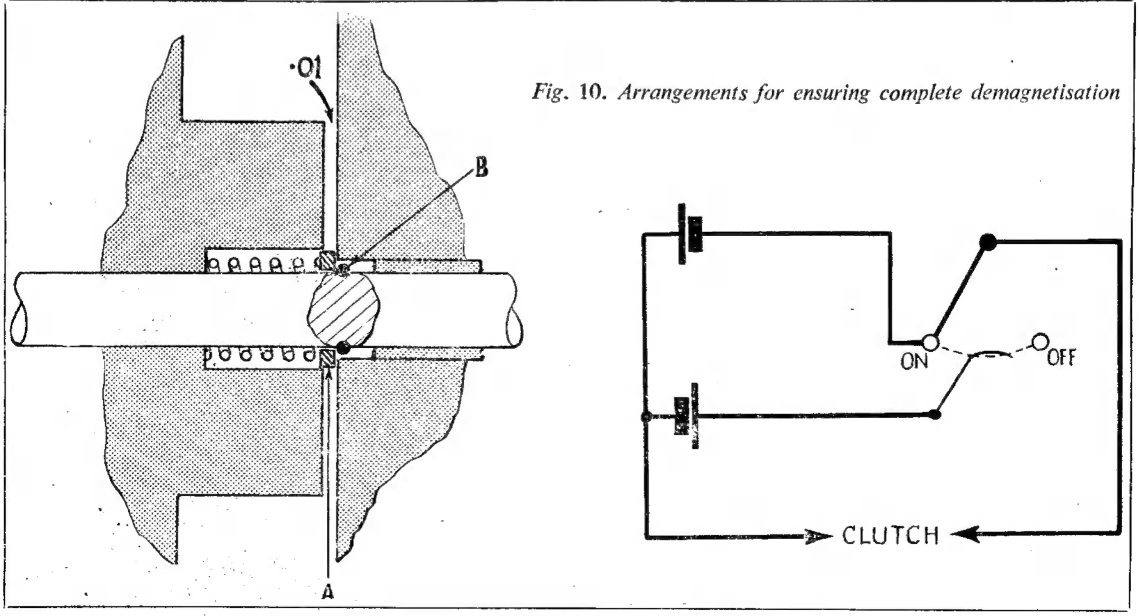


Fig. 10. Arrangements for ensuring complete demagnetisation

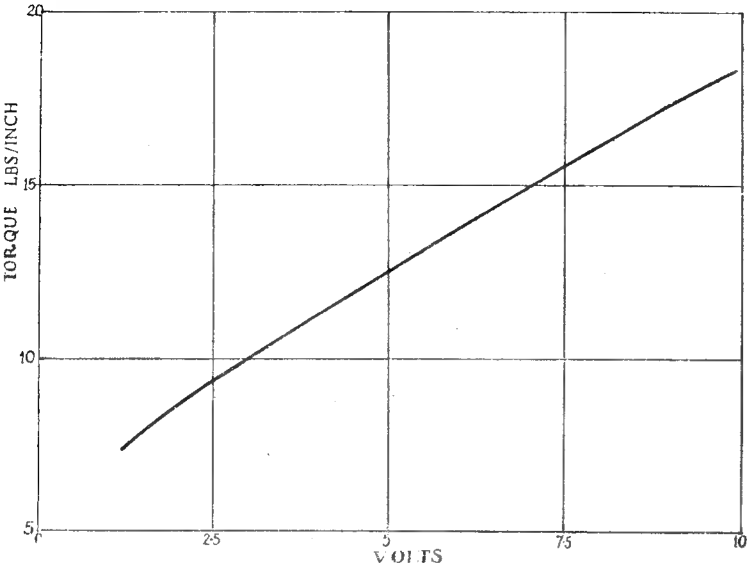


Fig. 8. Torque plotted against applied E.M.F.

accurately, but in general had a value of 1.25-1.5 times the sliding value. The latter value could be read fairly easily, and is plotted against volts in Fig. 8. It will be seen that the graph is nearly a straight line, but it must be remembered that the electrical input is increasing as Volts^2.

The torque value corresponding to an M.M.F. of 35.25 A.T. (as calculated), is, from Fig. 8, about 8 lb./in., and is thus considerably better than expected.

In view of the very favourable power input requirements, it was thought that a suitable power supply would be a No. U.11 torch cell.

This has a diameter of 1 in., is $1\frac{7}{8}$ in. long and weighs $1\frac{1}{2}$ oz.

A new U.11 was connected to the clutch for the second test, and checks made on voltage and torque against time. The result is plotted on Fig. 9 which clearly shows that the torque values are quite useful even after several hours' continuous running.

The general conclusions from the tests recorded were that a magnetic clutch is (electrically speaking) a thoroughly practical proposition, while mechanically it is extremely simple and robust. One snag, however, remained to be solved. It was stated that the remaining magnetic effect after cut-off of the current input was fairly constant at about $2\frac{1}{2}$ lb./in. torque. This effect was anticipated and its unvarying character is due to the fact that it depends upon the nature steel and dimensions of the clutch rather than the applied M.M.F. (provided the latter exceeds a certain minimum value).

This effect is, of course, common to all electromagnetic equipment and is usually solved by maintaining a small air gap somewhere in the system. This would be impractical, however, in the present case, but it is essential to remove all magnetic pull to enable the clutch to disengage fully. Ideally the working faces should be separated by a slight of the gap when the clutch is out.

The best method of achieving both these objects is shown in Fig. 10. On the left is shown a section through the hub of the unit ; within the counterbore is a compression spring butting at the left against the driving member, and pressing washer **A** against the circlip **B**.

The face of the washer is about 0.01 in. proud of the polar faces of the driving member, and, as shown, the clutch is disengaged with the driven half loose on the shaft and butting against the washer. The force of attraction between the two parts due to *permanent* magnetisation is, of course, negligible with mild-steel.

When the clutch is energised, the driven part is axially attracted and overcomes the force of the spring to move home against the driving part. After the current is cut off the driven half remains in position until the remaining magnetisation is cancelled. (To increase the strength of the spring in order to release the driven part would, of course, seriously reduce the working efficiency of the clutch and might prevent the two halves coming together when re-energised.) In this case, cancellation is effected by the simple circuit shown on the right of Fig. 10. The on-off contact for the clutch (which would probably be a selector) is arranged to brush momentarily against a connection as it moves " off." This connection is supplied with 1.5 volts in opposite polarity to that of the main supply.

The current in the clutch—which is a highly inductive circuit—takes

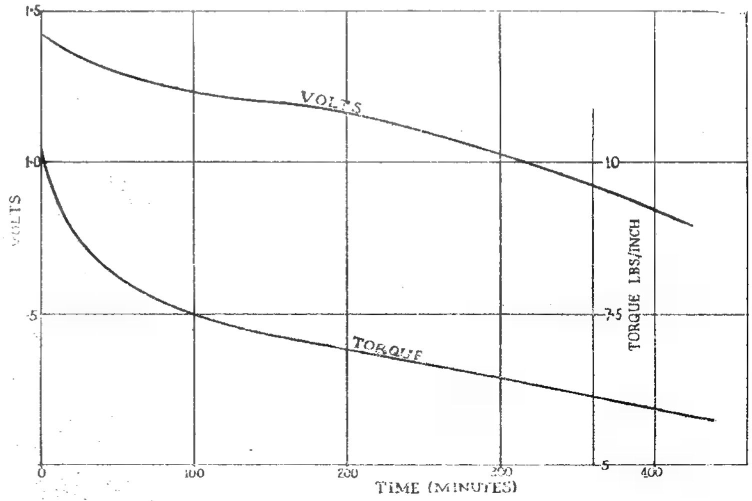


Fig. 9. Decay in voltage and torque when using a single U.11 cell

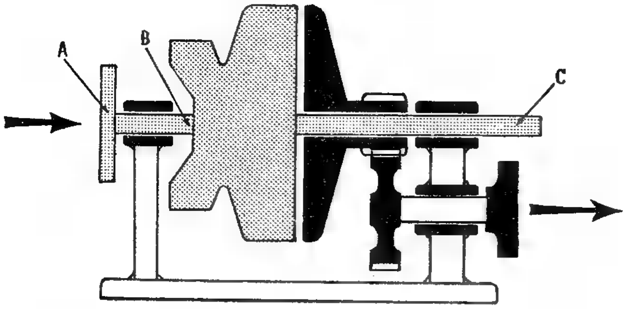


Fig. 11. *Proposed application of the experimental clutch*

an appreciable time to change from zero to maximum and vice versa. It is, therefore, easy to adjust this connection so that the reverse current is applied long enough just to neutralise the remaining magnetic force in the clutch. In fact, it is not necessary to carry out a precise adjustment, since as soon as the attraction diminishes to a certain low value the spring separates the two parts and thus produces full disengagement.

The spring in use at the moment consists of five turns of 20-gauge wire (this has a measured force of 2½ lb. when the driven part is fully home), but this is a point obviously best decided by experiment.

The clearance of 0.01 in., fixed by the position of the circlip groove is ample to separate the two parts and should not be exceeded, as it is important that engagement is quick and positive.

The system outlined depends on a slight time lag, but this is of a very small order and it will be found that the clutch can follow the average selector quite happily, considering its purpose.

The components of Fig. 10 are only one way of doing the job. When time permits, I intend to try the use of a condenser discharge circuit to apply a metered quantity of energy for depolarisation, and thus avoid the fleeting contact. Perhaps other readers may develop a better method.

Applications

So far, time has not permitted the complete testing of the clutch in a model under working conditions. The experimental unit has, however, been bench-tested on a motor and has done everything expected of it (including stalling the motor, etc., when overloaded) so no snags are anticipated when it is put into service.

The clutch is intended for use in a unit separate from the power plant, which latter would have its own (small) flywheel. Fig. 11 shows the projected design. Flexible drive **A** receives the input and drives shaft **B** to which the driving member is keyed. The driven member carries a pinion meshing with a wheel on the output shaft ; this arrangement has three main advantages. The propeller shaft is kept low and also runs at half engine speed, while a constantly rotating shaft is available at **C** to drive auxiliaries such as oil, fuel and water pumps. This is important when using a " single-ended " engine.

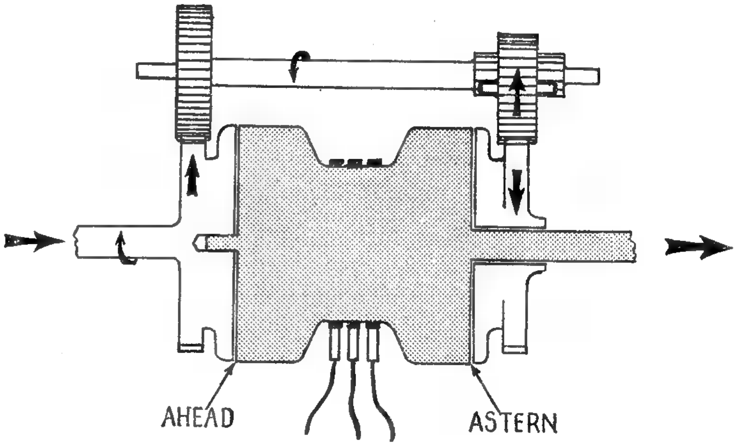


Fig. 13. *Projected idea for use in a reversing gearbox*

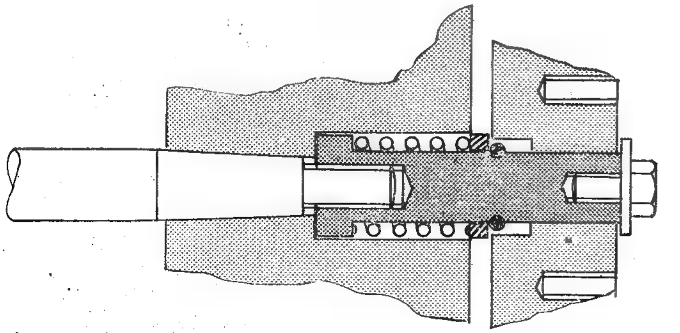


Fig. 12. *Suggested hub details for engine-mounted unit—not to scale*

For use with a simple power plant with no auxiliaries and where a high power/weight ratio is essential, the clutch adapts quite well to the layout used by the model car enthusiasts for centrifugal clutches. Fig. 12 shows the idea ; the bore of the driving member is tapered to suit the crankshaft and pulled up on to it by the stub-shaft acting as a nut.

As for future applications, there is no reason why two magnetic clutches built back-to-back could not be used to provide " Ahead "-" Neutral "-" Astern " in a marine-

(*Continued on page* 161)

MORE "HOT AIR"

By Wm. Richardson

IN this hectic age of speed and progress of "jet" engines, and the anticipation of "atom" power, it is refreshing indeed to find that there are still a few, who are out to preserve and champion the older, and slower-moving prime movers.

Although at heart a steam fan, like our old friend, B.C.J., I, too, am particularly fond of the almost forgotten, hard working, reliable and fascinating hot-air engine, which I feel convinced played a more important part in the evolution of power than is at first realised.

In the early nineteen-hundreds, this useful machine kept the wheels of small industry turning in all sorts of places, when other sources of light power were not available, and was used for pumping water, driving fans, laboratory mixing-machines and other devices, generating light *via* the petrol-air-gas plants, and many other tasks. Only a few weeks ago, I read in a daily paper, that a new(?) power unit had been developed abroad for use in the hot and very isolated districts, where it was not possible to install other types of engines. This was not some form of jet, or some new discovery in the "atom" line. Oh no! Just our old friend the hot-air engine, adapted for firing by the rays of the sun through suitably focussed magnifying glasses. A sober thought for 1953.

The Heinrici engines mentioned by B.C.J. seem to have been manufactured in an unlimited number of sizes, and were of superb workmanship. Most, though not all of these, had the power piston and the displacer in tandem, a neat, but complicated arrangement, which cut down to a minimum the distance the air had to travel, and also offered no restriction to its movement, owing to the displacer and power cylinder being one continuous tube, thereby calling for no connecting pipes.

Some time ago, I managed to secure one of these engines which was very much the worse for wear, and is probably about fifty years old. The

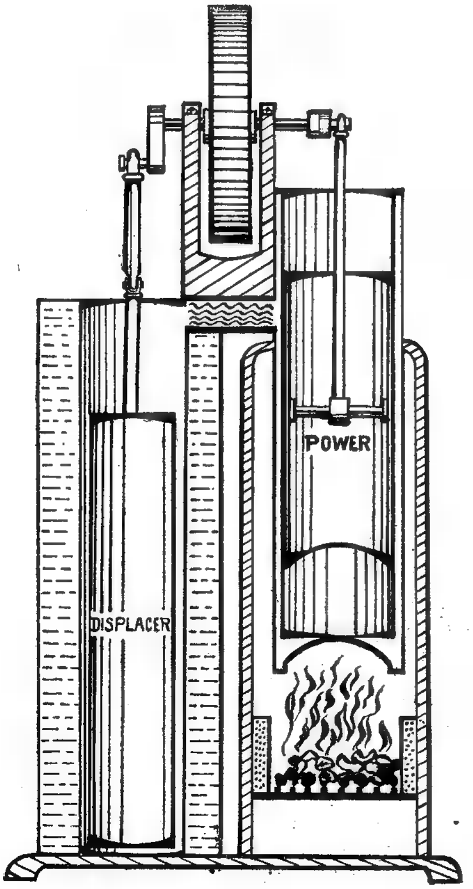


Fig. 1. *Showing the general design and principal of the Ryder hot-air engine*

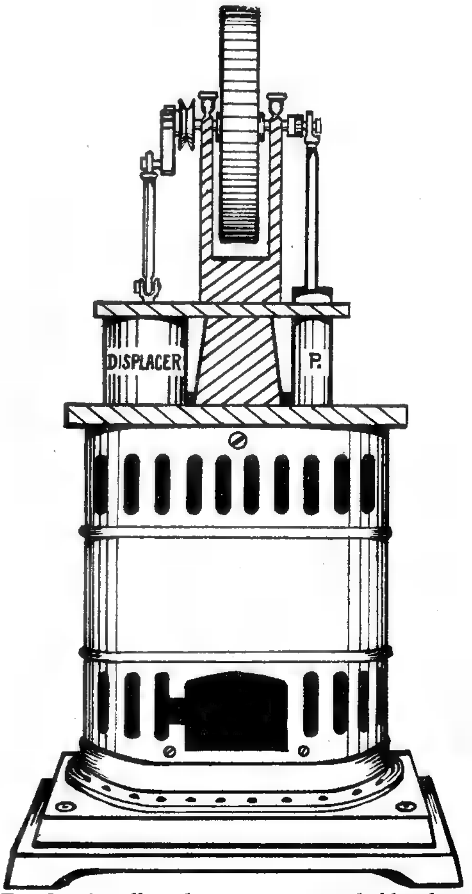


Fig. 2. *A well-made toy engine, probably about 35 years old, which bears a striking resemblance to the Ryder hot-air engine*

flywheels are only $6\frac{1}{2}$ in. diameter, with a face of $\frac{3}{4}$ in. The power cylinder has a bore of approx. $1\frac{3}{4}$ in. More like a glorified toy. Yet this dimunitive power unit, had actually been operating a four or five-mantle, petrol-air-gas plant.

After the renewal of several parts, this fascinating engine will operate quite well on a one-wick methylated spirit burner. I may mention, in passing, that in its working days, the engine was not cooled by water, as is the usual practice. The air, from the blower which the engine operated by a round belt, circulated round the water jacket and performed a dual function, that of cooling the air in the cylinder and also providing a suitable mixture for the gas plant. Another interesting point about these ingenious installations, is that once the gas works was in operation, the engine was fired by a jet of gas produced by its own efforts.

Another well-known make of engine working upon the Stirling principle was the "Ryder." Made in three standard sizes of $\frac{1}{4}$, $\frac{1}{2}$ and 1 b.h.p., this machine had a simple layout, although the orthodox method of applying the heat under the displacer cylinder and adopted by most manufacturers, was not incorporated in this design, the fire being directly underneath the power cylinder.

This is clearly shown in Fig. 1, which I sketched from an old text-book many years ago.

The power cylinder in this case is the larger of the two, and takes the form of a pendant liner open at both ends, the top to atmosphere and the bottom into the closed heating chamber. Within this liner moves the power piston, being forced upwards by the air which has been transferred from the water-cooled cylinder by the descent of the displacer, into the heating chamber, where it rapidly expands and increases the pressure throughout the whole system. However, this only acts upon the power piston, of course, as the displacer is not a piston in the true sense of the word, there being sufficient room between it and the cylinder walls for the free passage of air. When the power piston reached the end of the upward stroke, the displacer was half way up, therefore the subsequent descent of the former, transferred the air from the hot power cylinder, *via* the regenerator in the connecting pipe (where some of the heat was stored, ready to be absorbed during the next power stroke) into the cool displacer cylinder where the temperature and pressure consequently falls. Thus the cycle of operations commenced again, and was repeated as long as the fire continued to burn.

It seems to me that the design of this engine is admirably suited for a model, as the displacer and its cylinder could be silver-soldered, or possibly soft-soldered instead of being brazed, seeing that it is water-cooled. However, the bottom of the heating chamber would certainly require brazing.

Anyway, Fig. 2 shows an engine very similar to the general outlines of the Ryder, but following the usual practice of applying the heat under the displacer cylinder. This is a drawing of an actual toy air-engine in my possession, which I was amazed some time ago to see actually working on a junk stall in a small town market. The overall height of this well-made engine, which is probably 40 or more years old, is 12 in.

Some time later, I found another splendidly made engine of the horizontal type lying battered and broken in a back garden. Obtaining this wreck, I promptly stripped it down and eased out the kinks in the displacer cylinder with a turned wooden plug. After making some temporary new fitments, I was delighted to find that the engine runs really well. And that after searching for about fifteen years for a toy air-engine to carry out some experiments.

As B.C.J. remarked in his article in the May 14th issue (1953), there is considerable interest shown in the hot-air engine, and its followers should have a little space and playground in which to air their views and show their goods. Possibly in the distant future, at some exhibition or other, a few of us "slow crank-watchers" may be able to display our different engines in motion and all together. Then we should see where the crowds were.

AN ELECTROMAGNETIC CLUTCH

(*Continued from page* 159)

type gearbox. A typical scheme might be as illustrated in Fig. 13.

No investigations have been made into the use of the clutch under slipping conditions or the rate of wear caused thereby—but one is prompted to think of a R/C tank powered by, say, a "Seal" engine, and with a clutch to transfer a variable drive to each track.

The data recorded seems fairly reliable for an "average" M.S. sample. The efficiency of the clutch in terms of torque transferred for a given input depends (as in any electromagnetic appliance) upon how highly it is rated. In this case the overall size was rather overestimated and as made the clutch could, by using the appropriate input, easily handle the power output from larger engines of 10 c.c. or so.

Any readers wishing to utilise the idea for the smallest engines could scale down the proportions with every chance of success ; one of the advantages of this clutch is that it could be applied successfully to a power unit of any size, even as small as 1 c.c.

The winding can, of course, be adapted to suit any existing voltage by choosing a suitable wire gauge. An extract from the copper wire tables (enamelled) printed here may assist.

The brushes in use at the moment are $\frac{1}{4}$ in. $\times$ $\frac{1}{8}$ in. wide and are cut down from standard electric motor brushes of a soft quality. The wear should be negligible, so that arrangements for holding the brushes need not have provision for frequent changing.

In any practical installation the brushes could be carried in rectangular tubes buried in a block of Paxolin, and this would best be arranged to one side of the slip rings to permit the clutch to be mounted low in the hull. If a vee-groove is used for starting, as shown, the brush block would require to be substantial to avoid damage from the end of the starting cord.

Gauge	Res. ohms/yd.	Turns sq. in.	Turns per in.
26	0.09448	2,560	50.6
28	0.1398	3,760	61.4
30	0.1991	5,370	73.3
32	0.2625	6,890	83.0
34	0.3617	9,610	98.0
36	0.5300	13,500	116.0
38	0.8503	20,400	143

Cutting cylinder latch keys

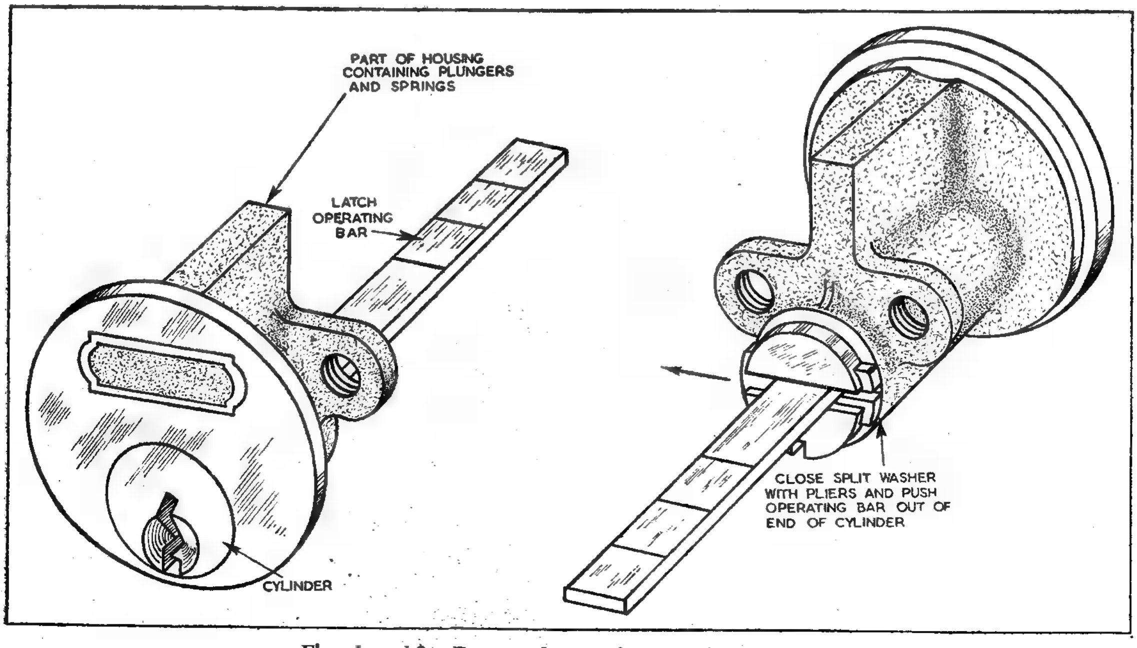
By S. E. Capps

WHEN the only key to the front door lock is lost, and it happens to be one of the cylinder type, it usually means a new lock, as keys made locally for these locks can only be obtained to an existing key. New locks of this type are, like everything else today, expensive. Having collected, over a period, several of these cylinder locks in good condition, I decided to dismantle one and cut a key to suit it. Now the internal mechanism of these locks is not the complicated mystery its outward appearance would suggest, and to dismantle one and fit a new key is well within the skill of the homeworker used to small metalwork. The procedure I adopted is shown in the accompanying sketches. There are a number of different makes available, but the general internal design is the same. Figs. 1 and 2 show the front and rear views of a typical cylinder lock with the latch operating bar. Fig. 3 shows the locks in section. Here it will be seen that sets of round plungers are held down into a revolving cylinder by small springs contained in the housing above them.

It will be observed that these plungers are of different lengths, and it will be understood that as the key is inserted into the cylinder it raises each bottom plunger to the outside diameter of the cylinder, which can then be revolved inside its barrel, turning the operating bar with it. This turns the latch bolt, which is withdrawn from its staple on the inside of the door frame, and the door can be opened. A spring inside the latch casing forces the bolt out again when the key is released or withdrawn. When the key is withdrawn, the plungers drop back into the cylinder, and lock it in place until the key is re-inserted.

To dismantle the unit, remove the rear operating bar by removing its retaining-screw or split washer. These bars either slide out of a slot in the end of the cylinder, or take straight off the end. The cylinder, in its locked position, cannot be withdrawn from the housing. It will be obvious that before the cylinder can be freed, the plungers must be raised either to the right height or out of the housing altogether. The latter is probably the only way for the homeworker. Referring to Fig. 3, file away the top of the housing containing the springs and plungers individually as shown, until the end of the plunger hole appears. When near the full diameter remove the last bit of metal with a twist drill. Extract the spring, and top and bottom plungers. Place these in a groove filed in a block of wood prepared for them as shown in Fig. 4. Keep them in the right sets, and in order of extraction.

When all are out, reverse the positions of the longest and the shortest on the block, just in case someone has a key of your door who shouldn't. The cylinder is now free and can be withdrawn. This should be cleaned and blank keys to fit its centre aperture obtained from the local ironmonger's store. Fit a blank to the cylinder and slide it into the housing. It should slide in and out freely. See that the shoulders of the blank fit up tight to the cylinder face. Withdraw the cylinder and key and secure the blank in the cylinder with soft wire, as shown in Fig. 5. Hold the cylinder



Figs. 1 and 2. Front and rear of a typical cylinder lock

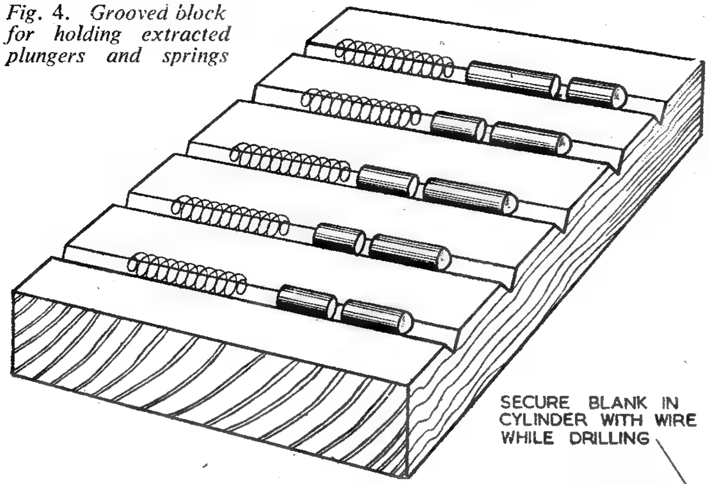


Fig. 4. Grooved block for holding extracted plungers and springs

between lead clams in the vice, and with a drill that fits the plunger holes held in the wheel brace, drill down into the blank carefully, a little at a time, until the bottom plungers are just flush with the top of the holes. Note in Fig. 5 that the combined cutting angles of the drill are somewhat flatter than the normal angle of twist drills. Carry out this operation very carefully, as, should one plunger be below the top of its hole, the blank is scrap as it will not work when reassembled.

On no account should the bottom plungers be above or below the outside diameter of the cylinder. If one only is above or below, the cylinder will not turn in its barrel. Try the cylinder as each plunger is fitted. When all the plungers are correctly fitted, withdraw the blank from the cylinder. This may jam a little, due to the burr caused by the drill, but a little moving to and fro will free it. Remove the tops of the intervening metal between the plunger positions on the key, until it slides in and out of the cylinder smoothly. Fig. 6 shows the easy rise and fall of the plunger positions. Put the bottom plungers into the cylinder, with the key in place, and slide the cylinder into the housing. Hold this in tight and drop in the top plungers, followed by the springs. Place a piece of metal over the springs and compress them down tight. Now test the key for turning. If it sticks, one or more of the plungers are not quite correct, and this must be rectified.

When correct, withdraw key and test for locking. If the key unlocks the cylinder, and it re-locks on withdrawal of the key, the positions are correct. Refix the operating bar on the rear of the cylinder, and solder a piece of brass to the housing, over the springs, to keep them in place. This completes the job. Fitting these keys is a lot easier than it looks, and makes an interesting as well as a money-saving job. Once you have one satisfactory key, get some more cut from it at the ironmonger's. Repeat keys are cheaper obtained this way.

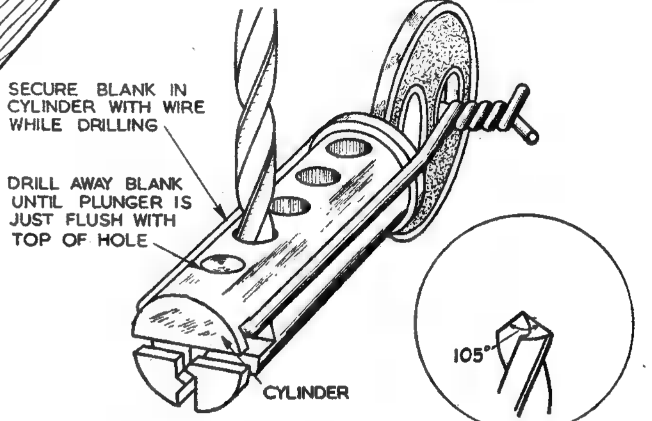


Fig. 5. Drilling the plunger positions in the blank key

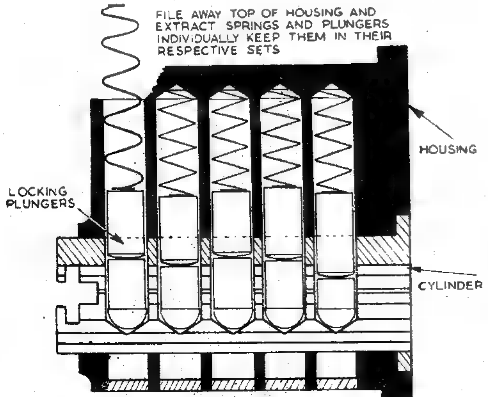


Fig. 3. Sectional view, showing the locking plungers

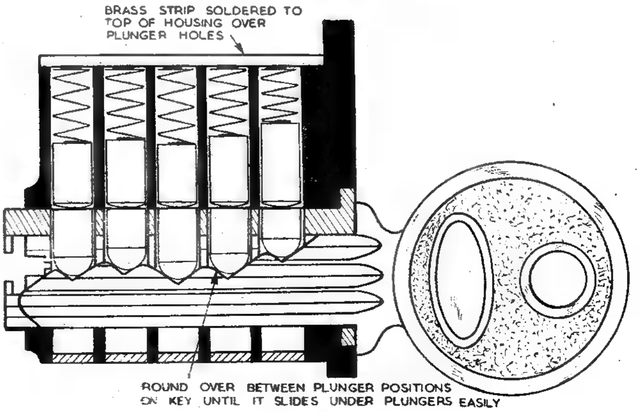


Fig. 6. Finished key holding plungers in the unlocked position

LAYOUT OF STEPHENSON LINK-MOTION

Notes on planning the Valve-gear for a ¾-in. scale G.W.R. Bogie Single Locomotive

By D. G. Webster

THE subject of laying out Stephenson's link motion has been dealt with in the past, especially by Mr. G. S. Willoughby in Vol. 77, Mr. Dunn in Vol. 96, Mr. J. N. Maskelyne in Vol. 80 and several times by that master of miniature locomotive construction "L.B.S.C."

My own humble effort is to show a practical application to a ¾-in. scale G.W.R. "Dean" single, which I am constructing. This engine is intended to be a close reproduction of No. 3031, *Achilles*, at a certain period of her life, using the official G.W.R. general-arrangement drawing together with other information in my possession.

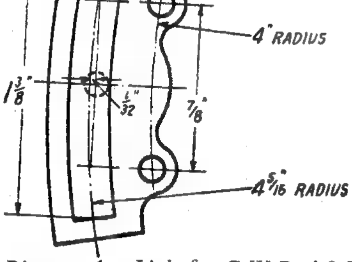


Diagram 1. *Link for G.W.R.* 4-2-2

For practical reasons, if a sound working model having long life to its moving parts was to be produced, some departures from the prototype had to be made, at the same time preserving the characteristic appearance of the original.

The singles had double frames of ¾ in. and ⅞ in. thickness, and the inside frames had two set-ins to give clearance to the bogie. The main motion inclined 1 in 33 and the valve motion declined 1 in 12 to the driving centre.

The cylinder block casting contained the cylinder, valve-chest and back-cover with the slide-bar brackets and piston-rod gland housing carried by a smaller auxiliary back-cover. The valve-chest also had removable end covers.

At the point where the cylinder-block is located, the inner frames were 3 ft. 10 in. inside, with the cylinders at 2 ft. 3 in. centres and 19 in. bore × 24 in. stroke; slots were cut in the frame to allow the block to protrude through the frames slightly, the cylinder-block was installed at the time the frames were erected.

The connecting-rods were 7 ft. between centres, both ends having strap and gib-and-cotter type bearings, the crosshead bearing being double.

I intend to build the frames up exactly as the originals, with all the brackets and stretches riveted together. With a little thought given to sequence of erection, this is not too difficult.

The frame thickness was settled at 5/64 in. as the minimum that could be used with reasonable rigidity, and to preserve the important outside dimensions this added thickness has to come off the available inside dimensions. It is here that the first modification was made by altering the cylinder-block. While the block could be made exactly as the original, especially if fabricated, it is simpler to make both covers and valve-chest removable; also, if the bore is made a scale 19 in. to keep the overall diameter of the covers to scale would not allow a very satisfactory joint to be made at the cylinder ends, and owing to increases in metal thickness, to do this the cylinder lagging would protrude too far and involve alterations among other things to the smokebox brackets. Finally, to get the cylinder-block out or do any jobs on it would be difficult with the frame construction I want to use.

By reducing the cylinder bore to 1 in. or 1$\frac{1}{16}$ in. maximum and making the cylinder-block to slide in after frame erection, these problems disappear especially as cylinders of this bore will give me all the power I want to develop with the engine, while the main features, bore centres and outside appearance remain unaltered.

As stated previously the cylinder-block, valve-chest, etc., will be made up separately, but by the time I had laid it out the centre of the valve-motion declination had increased from 1 in 12 to 1 in 9. This came about partly because I wanted to cut out as much as possible any sharp bends in the steam-port passages, and partly owing to a modification to the valve-rod.

In the original, the valve-buckle is formed solid with the rod, the buckle being set up about 1 in., i.e. the rod just clears the port face. Mine, however, is made with the valve-rods secured central to the buckle. The other type *can* be made but the latter is simpler.

By this time, I had realised that it was essential to draw a full sectional arrangement through the engine to check that everything else would go in. and I was relieved to find that, except for two points involving the valve-gear, everything could be made to scale with due allowance for strength, etc.; but before mentioning what these were, some other points may be of interest. There have been several arguments in the past as to what cut-off should be used in a miniature locomotive. My own view is that this should be somewhere about 80 per cent. of piston stroke in *full gear*, especially in the case of large-wheeled engines of small adhesive weight. A single-wheeler comes into this class.

Unless this is done the possibility of slipping is increased, due mainly to the crank turning effort fluctuating considerably above and below a mean in one revolution, coupled with low revolutions of the driving wheels, and this at a time when the resistance to movement of the load is greatest. I understand that to see a "Dean" single pulling out of Paddington, in the hands of a driver who *knew* his engine, was something worth seeing. One can still see something like it today at the same station. Watch a driver leaving No. 2 or 3 Platform with a heavy train, and the method is nearly always the same. Nowadays, at least two carriages and the engine will be on a sharp curve. The engine will be well towards, if not actually in full fore-gear. At "Right Away" the driver gives the throttle a good pull, and the engine moves

smoothly out; no slip, no fuss. As soon as the engine has turned a few revolutions the driver is linking up without any further adjustment to the throttle. It is then that one realises the skill of designer and driver, when both know their job.

I know G.W.R. engines can and do slip, but it is not normal. One thing I will say: they are soon checked, not like some of the other regions! This is not necessarily the driver's fault.

From this little story can be seen that, although for starting, it is better to have a late cut-off, even with a model it should be linked up when under way with the reservation that a model has to work on both continuous and up-and-down tracks while the full size express locomotive spends most of its running time travelling forward only.

The " Dean " singles had locomotive link gear suspended at the fore-gear end. This will give good valve events in fore-gear, but back-gear valve events leave something to be desired, due to the distance from the point of suspension; also link slip increases on back-gear for the same reason. I say " link slip " because a fixed die-block cannot slip.

A centre-suspended launch-type link does not suffer these difficulties so much, and to improve the valve events for the two types of track, I decided to modify the valve-gear to launch type. I also wanted to take advantage of a long lap to increase the valve-speed and reduce wire-drawing and the increased travel would have meant increasing the size of the eccentrics and the overall length of the link to avoid the angular swing of the link exceeding the accepted maximum of 30 deg. Finally, with end-suspended locomotive links, the suspension arm-pin must be free on both eccentric-rod pin-holes and the link, and reasonable wearing surfaces could not be got in with the double crosshead bearings, a feature I want to keep. This does not arise with launch type links, centre-suspended, and the eccentrics are not much larger in diameter than the original with the valve travel increased to 13/32 in. ($6\frac{1}{2}$ in.).

The port sizes and valve travel were decided from the following data, all to the nearest common dimension :—

Port width=1/10th cylinder bore. Port bar width=port width. Exhaust port width=2 × port width. Length of all ports=$\frac{3}{4}$ cylinder bore. Valve lap=port width.

Full gear travel for given cut off, can be stated thus:—

Steam lap × 4 = 75 per cent.
„ „ × $4\frac{1}{2}$ = 80 per cent.
„ „ × 5 = 85 per cent.
for mean cut-off at both ends

This formula assumes there is nil lead, and for ■ cut-off greater than 75 per cent. the valve over-travels the exhaust edge of the port, in the case of 80 per cent. cut off setting $\frac{1}{4}$ the lap, and in the 85 per cent. setting $\frac{1}{2}$ the lap. This is not a very bad thing, but does involve the gear in doing a little extra work.

In my own case, the lap was made 3/32 in. with 7/64 in. ports and nil lead. This gives a full-gear travel of 13/32 in. without over-travelling the port and a mean cut off of 78.5 per cent.

There is a point about this that is not always appreciated. The above cut-off is obtained by the given lap, lead and valve-travel set to an associated *angle of advance* of the eccentric. If the eccentric is retarded the lead will become negative and the cut off, exhaust opening and closing take place later, while the lead will become positive and all the other points occur earlier if the eccentric is advanced. If the eccentrics are made adjustable on the axle this is an advantage as long as it is remembered that it still holds good when the gear is linked up.

Having decided on the valve-travel the link was proportioned from the following: Distance between eccentric-rod pin-hole = Full gear travel × $2\frac{1}{8}$. Note: This is full gear travel × 2 for eccentric-rods over scale 4 ft. 6 in. length. Length of curved slot = Eccentric-rod pin centres × $1\frac{5}{8}$. Motion-pin diameters = $\frac{1}{8}$ piston diameter. Distance from centre of curved slot to centre of eccentric-rod pin-holes = Pin diameter × $2\frac{1}{2}$. Width of curved slot = Pin diameter × $1\frac{3}{4}$. Thickness of metal in front of curved slot = $\frac{1}{2}$ width of slot. Diagram (1) shows the link drawn out to the appropriate sizes for my own engine.

It is now possible to make the valve-gear layout drawing to find the best position for the link suspension pin, the hanger arm and bell-crank arm lengths, and also the amount of link slip that can be expected. While the method of finding the lengths has been described before, I have never seen an actual example of the link slip shown graphically for Stephenson link gear given in THE MODEL ENGINEER. Diagram (2) shows the method I used. The original is drawn to a scale of twice full size.

A template of the link was drawn on a piece of hard thin cardboard similar to Diagram (1). On this was marked: (1) The horizontal centre-line of the link. (2) An arc equal to the length from the eccentric

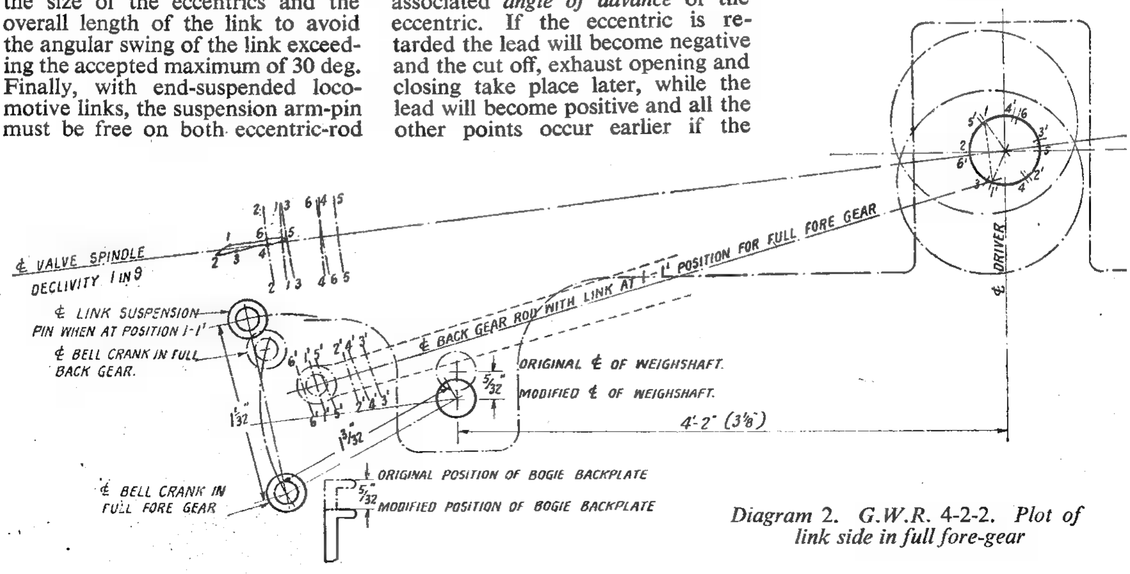


Diagram 2. G.W.R. 4-2-2. Plot of link side in full fore-gear

centre to the eccentric rod-pin centres, also to the vertical centre arc of the link curved slot.

The eccentric centre is found by drawing a circle at the driving centre equal in diameter to the valve travel (in the case of launch-type links). Draw lines from the axle centre at the appropriate angle of advance to cut this circle. If a connecting line be drawn to join the two cutting points where this line cuts the valve-motion horizontal centre-line is the point from which the arc is struck. Again, with launch-type links, it is not necessary to mark off the angle of advance; if the amount of lap and lead be marked off parallel to the vertical centre-line of the valve-motion at the driving centre it amounts to the same thing. Should it be suspected that this radius will give too early a lead, etc. in linked up positions, the link and eccentric-rod pin arcs can be struck from the driving centre. Both points given are correct, the first following normal British practice while the second was sometimes used in American practice.

(3) Draw horizontal centre-lines from the eccentric-rod pin holes to cut the vertical centre-line arc of the curved slot. This gives the centre of the die-block pin in full gear.

(4) Draw a line through the die-block pin centres which, dependent on the link radius, will cut the link horizontal centre-line somewhere behind the vertical-arc. The point at which this vertical centre-line meets the horizontal centre-line is the foremost point at which the suspension-pin should be positioned, and where the back edge of the curved slot crosses the horizontal centre-line is the farthest back the pin should go. If a point be taken half-way between them a good all-round result will be assured, but a little trial and error will show the exact point.

Draw the valve-gear out with the link first in mid-position and, with a slide-valve and open rods, the crank on back centre with the eccentrics on their common fore-centre. Draw the bell-crank parallel to the valve-centre line, and draw the hanger arm vertical to these centre-lines, passing through the mean point of suspension. To avoid errors due to arcing, the bell-crank length from weigh shaft centre to the longer pin-centre should be at least ¼ the eccentric-rod length and the hanger arm as long as possible.

Draw the bell-crank in full fore-gear position, and prepare a template of the hanger arm pin-centres, and pin one end of it at the bell-crank pin-centre. With a drawing pin, head downwards, fix the link template to the other hanger arm pin-centre, having made sure that either by cutting notches or holes that the eccentric-rod pin and die-block pin centres can be sited through on to the drawing.

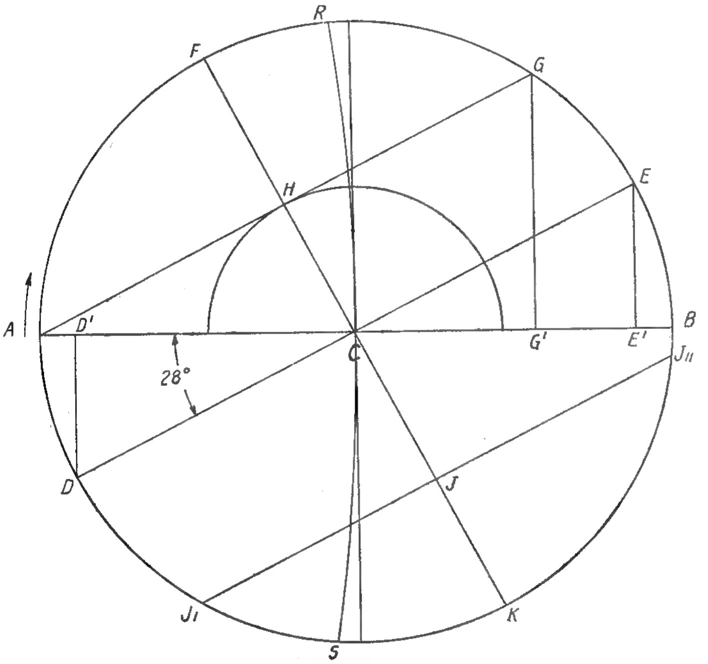


Diagram 3

Divide the valve-circle up, first using the fore eccentric centre, then the back eccentric centre, and mark them for identification. From these points strike a series of arcs with the compasses set to the length of the eccentric-rod. The fore-rod arcs fall across the valve-motion centre-line while the back-rod arcs are struck roughly in the position occupied by the back eccentric-rod pin. Mark these and if the link template is now swung to each appropriate pair of marks and a mark made for the position of the die-block pin-centre the amount of link slip is obvious. It is now that trial and error comes in, as adjustments to the point of link suspension and slight alteration to the relative position of the bell-crank pin can make quite a lot of difference to the amount of link slip, and where it occurs. This is important, not only from the point of valve event errors, but also wear.

Finally, having got the best possible pin centres, check everything in back gear to see the effect there. In my own case, the centre of the weighshaft was already fixed, its bearings being located in an extension of the frame.

As shown in Diagram (2) the alteration to the valve travel and link plus heavier sections involved two further modifications to the structure as follow :—

The heavier section of the eccentric-rod plus a different point of fixing, and the increase from 1 in 12 to 1 in 9 entailed dropping the weighshaft centre 5/32 in. to clear the eccentric-rods. I did not want to move this centre longitudinally as it is mainly hidden by the front footstep in its proper position, and if I went forward, the length of the bell-crank decreased and that is already short enough ; if I came back, I was spoiling the outside appearance.

The second modification involved the bogie back plate.

The original is shaped like an elongated " U " on its side, among other reasons to clear the valve gear and bell-crank arm. Again, owing to longer travel and heavier construction this plate had to drop 5/32 in. to clear the bell-crank in full fore-gear. None of these modifications is obvious, and it would take an expert of the calibre of Messrs. Maskelyne, Keiller or Dunn to spot that it has been done !

As to my method of testing the gear layout, while it is not quite

the orthodox method, it does give a simple way without going to the trouble of making a model of it which, if it is to give good information, must be well made.

Valve Diagrams

While valve diagrams do not necessarily show the actual valve events, they are useful to ascertain some idea of what to expect with a given setting. Up to now, whenever other writers referred to diagrams as applied to locomotive practice, they have always used a Zuener diagram. To my mind, this diagram is much more difficult to understand and construct compared with a Realeaux diagram, and text books on both locomotive and marine practice recommend the use of the Realeaux in preference to the Zuener. Construction to find the remaining data is simple with a Realeaux. It can be done for a Zuener, but it is very complicated.

Diagram (3) shows a Realeaux diagram and is the full-gear diagram of my layout for the "Dean." *AB* is the valve travel to scale, and also represents to another scale the stroke of the piston. *ACD* is the angle of advance. *CH* is the steam lap. *DCE* is drawn parallel to *AHG* and the crank is moving from front to back centre.

For any position of the crank, the piston position is given by dropping perpendiculars to *AB* while the corresponding valve positions are found by perpendiculars to *DE*.

When the crank is at *G*, the piston is at *G′* to scale from the beginning of the stroke, while the valve has moved the perpendicular distance from *G* to *DE* from its mid-position. As this perpendicular represents the lap, cut-off occurs at *G*. It is thus clear that steam is admitted at *A*. Maximum port opening occurs at *F* and cut-off at *G*.

The lower half of the circle shows the exhaust events and, as drawn, represents a valve with an exhaust cavity the same length as the distance between the exhaust edges of the ports, i.e the inner edges for outside admission. *E* is the point at which the exhaust starts to open and the exhaust remains open until *D*, when the valve closes. *CJ* represents the width of the exhaust port and from *J″* to *J′* the port is full open. From *D* to *A* the remaining exhaust steam is compressed. It can now be seen that release (exhaust opens) occurs at *E* and compression (exhaust closes) commences at *D*.

It is the normal practice to express the distribution points as percentages of the piston stroke. The perpendiculars on *AB* at *GG′*, *EE′* and *DD′* give the *mean* cut-off, release and compression points, respectively. I say *mean* as the diagram assumes a connecting-rod of infinite length.

In practice, this is not so. If a drawing be made of a cylinder of 2 units length, a crank of 1 unit length and a connecting-rod of 4 units length it will be found that, if the crank is drawn at half stroke the piston is *below* its half stroke in the cylinder. This is also true for the valve, but the relative length of eccentric-rod to eccentric throw makes any obliquity of little moment.

In locomotive practice, with connecting-rods of greater proportionate length to crank length this relative difference of half piston stroke to half crank stroke is reduced. It is still there, but is usually ignored. It can be and is corrected to a certain extent in vertical marine and high-speed engines, where, to keep the engine height down and cut out vibration, the connecting-rods are short, by making the bottom lap less (thus increasing the lead) but not necessarily to balance the cut-off.

If it is desired to find the correct points of cut-off, release and compression on the diagram, a radius *RCS* equal to the length between the connecting-rod bearing centres is measured to the same scale that *AB* represents the *piston stroke*. The measurement from *G* to *R* parallel to *AB* transferred on to *AB* then gives the true position of *G′* and similar treatment of *E* and *D* will give these points.

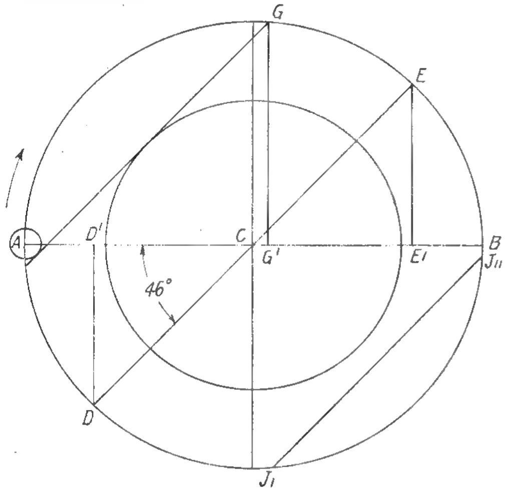


Diagram 4

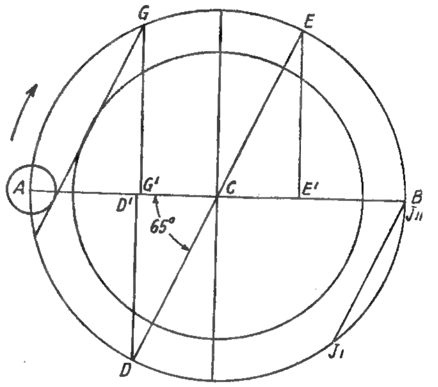


Diagram 5

Diagram (3) represents the piston moving from front to back centre and radius *RCS* is struck from an extension of *A*. If it is desired to find the true distributive points for the other stroke, *RCS* should be struck from the bend of the diagram.

To revert to the construction of Realeaux diagram, Diagram (3) shows nil lead and exhaust clearance. If, however, there is lead a circle of radius equal to that lead is struck with *A* as the centre. If the lead is positive, *AHG* is drawn as an internal tangent to the lead and lap circles, while if the lead is negative, *AHG* is drawn as an external tangent to the lap and lead circles.

Exhaust clearance is shown by drawing a line parallel to *DCE* at a distance equal to the clearance *above DCE*. Similarly, exhaust lap is drawn *below DCE*.

(*To be concluded*)

Traction-engine tea!

By W. J. Hughes

CORONATION Day was celebrated in many different ways in various parts of the country. At the village of Barlow, near Chesterfield, plans had been made for a "field day" in traditional country style, with a parade, sports, dancing, and a hearty knife-and-fork tea.

Unfortunately the sports and the parade had to be postponed because of the high wind and heavy rain, but the tea was a huge success, with mountains of ham and beef, salad, fruit, and cakes, all washed down with gallons of tea.

Now gallons of tea require gallons of boiling water, and in Barlow that calls for the assistance of "Ned" Morgan and *Dolly*, the latter being a Burrell traction-engine of 6 n.h.p., built in 1924. For this auspicious occasion she was furbished up until her paint glistened, her motion work was like silver, and her brass-work like gold; her stay-heads, rivet-heads, and cylinder-nuts were painted silver. Her full decoration with bunting and coloured paper was prevented by the weather, but flags flew bravely from her chimney base and cylinder.

Dolly's boiler had been washed out until the water ran crystal clear from the mud holes, and then washed out again just to be sure. The man-lid on top of the boiler was used for delivery of the water through a hose, and the boiling water was drawn off through the blow-off cock as required for tea-making and for "washing-up."

There was a great satisfaction to a traction-engine fan to be obtained in drinking tea thus made, and very good it was to have the opportunity. Despite the pelting rain, the huge marquee was full of conviviality, and certainly the weather did nothing to spoil the appetites of those present. Meanwhile, out in the rain, *Dolly* sang gently—almost beneath her breath—to herself, well-knowing the affection her owner bears for her.

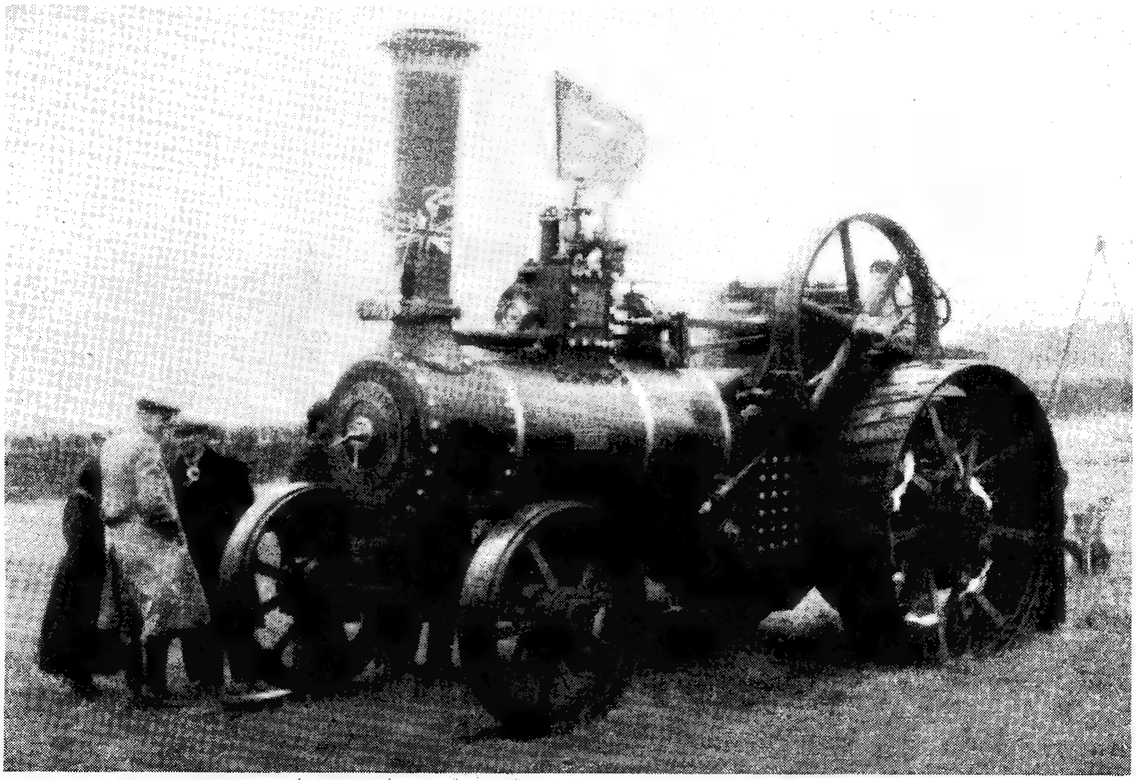
Since the Jubilee.

As a matter of fact, the Morgan family have performed this water-boiling service in Barlow for every Royal occasion since Queen Victoria's Jubilee in 1887.

The engine on that occasion was Fowler No. 4267, new in 1882, and she served again in 1897 at the Diamond Jubilee, in 1902 at the Coronation of King Edward VII, and in 1911 at that of King George V.

At the Silver Jubilee in 1935, and at the Coronation of King George VI, the engine used was Marshall No. 12218 (who incidentally celebrated her own half-century in 1936).

Now the Burrell No. 3984 has helped Barlow to celebrate the Coronation of the young Queen Elizabeth II. Let us hope that *Dolly* will still be there to provide the boiling water at the Silver Jubilee of 1978 !



" Dolly " poses for her Coronation picture while owner Ned Morgan (with rosette) chats to friends

A DIVIDING DEVICE for the "M.L. Super 7" Lathe

By A. R. Turpin

DURING the past year I have been doing a considerable amount of gear cutting, and found that, provided one possessed a separate milling spindle, the most convenient position for a dividing-head was on the mandrel of the lathe. With this arrangement, fitted to the rear of my M.L.7, I have done most of my gear cutting; the blanks being turned to size and then the teeth cut without removing the blank from the chuck.

The dividing-head that I used was described under the title "A Dividing-head, Plus," which was designed so that it could be brought out of engagement with the worm wheel by releasing one bolt only. Of course, I never replaced the gear case, because that would have meant removing the dividing-head, and, anyway, most amateurs seem to leave it off.

Recently, however, I took delivery of that masterpiece of Myford's their "M.L. Super 7." As I was still busy gear cutting, the first operation was to find a means of fitting my dividing-head to this lathe. In this case, the gear cover is a much more massive affair and is hinged so that the opening and closing of it is a matter of seconds, which entices the amateur to use this safety adjunct more than he would on other lathes. I had, therefore, to design a fitting that could be removed and replaced easily.

Photographs Nos. 1 and 2 show how this was done.

Actually, the cover will not close on this bracket without cutting some of the cover away, and this was my original intention; but when I had thought the matter over, I came to the conclusion that this would be a silly thing to do, because the bracket could be removed and replaced in about five minutes. Nevertheless, I thought that perhaps even this amount of trouble might stop my using the head for, say, filing up a square on a shank, or the like; and so I said to myself, "Why not have a second and simpler type of indexing device for this

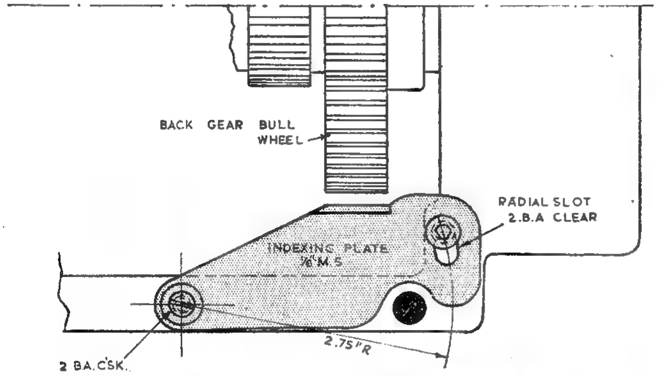
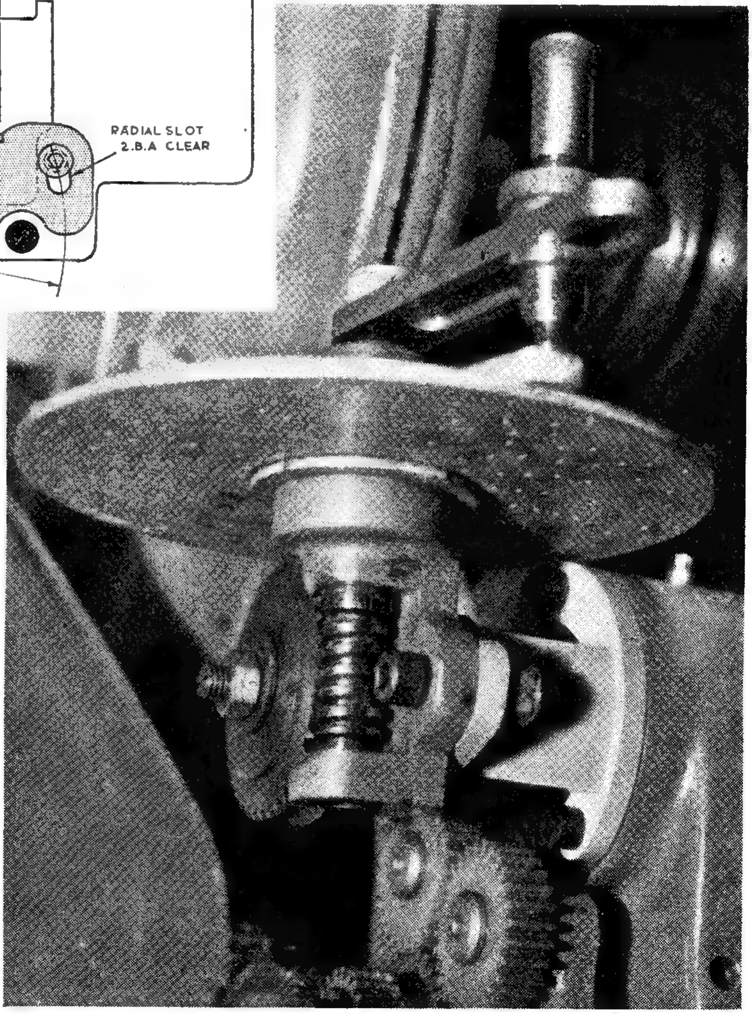
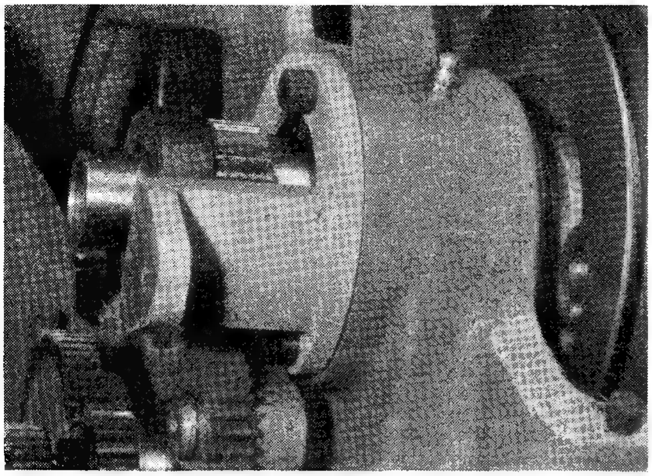


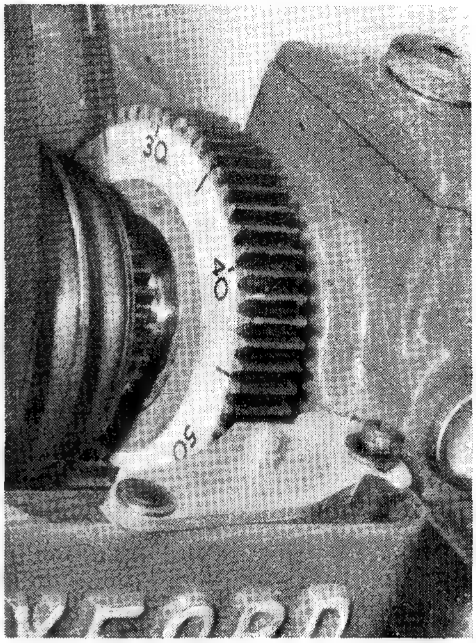
Fig. 1. Indexing device, using bull wheel teeth



Right—No. 1. How the dividing head was fitted to the "M.L. Super 7"



No. 2. *The bracket fixed to the back of the rear bearing housing of the mandrel*



Right—No. 3. *Using the bull wheel of the Super 7 as a dividing device*

simpler type of dividing : one that can be left in position, and is always available at a moment's notice."

The best arrangement for such a device would be to mount a plunger type of detent on the rear bearing housing, so that the point engages in a row of holes round the rim of the rear and largest of the belt pulleys ; and provided that I was willing to cut a small piece out of the belt cover, the function of the lathe would not otherwise be interfered with, and the only snag was that I would have to dismantle the mandrel. It was at this point that I noticed the bull wheel of the back gear ; if by chance this had, say, 40 or 60 teeth, half of my job would have been done for me. It *had* 60 teeth. Photograph No. 3 shows how this wheel was utilised, and all that had to be done so that the belt cover will still close was to drill a shallow hole in the bottom edge of the cover to take the head of the Allen screw. Now let us return to the original bracket for the dividing-head proper.

Photograph No. 1 shows the head mounted in position, and the worm engaged with the worm wheel tha is fixed to the end of the hollow mandrel by that old device, the expanding bolt.

Fig. 1 shows the drawing of this bracket, which can be cast in iron, or fabricated from mild-steel by brazing. The drawing is slightly different from that shown in the photograph so that the division plate will not be tilted away from the operator as it is in mine ; this is overcome by offsetting the lug on the bracket. The reversing tumbler has to be in the down position when fitting or removing, and there is so very little clearance when putting in the screws that I think those of the Allen type are almost a necessity. It will be seen that the dividing-head may be removed by loosening one

(*Continued on page* 172)

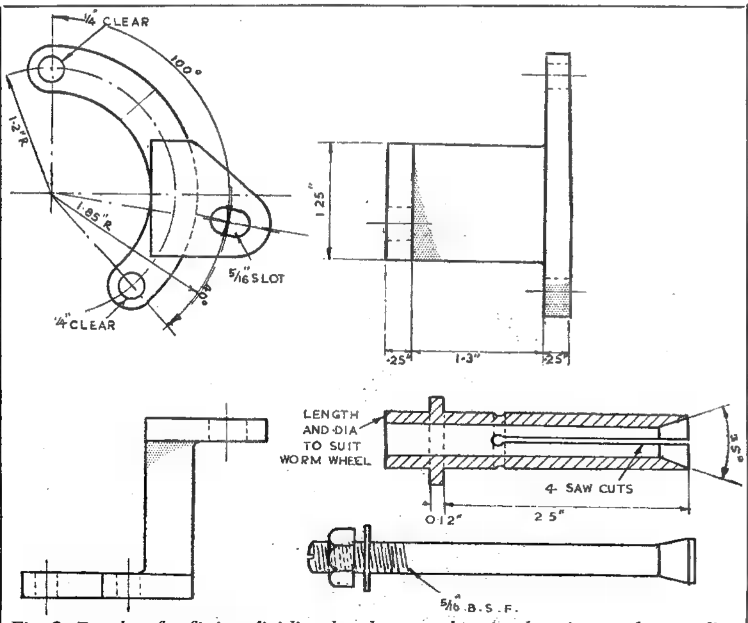


Fig. 2. Bracket for fixing dividing-head to rear bearing housing, and expanding bolt for fixing worm wheel to mandrel

Balancing a lathe countershaft pulley

By "Duplex"

MOST owners of small workshops, no doubt, get asked by friends from time to time for advice, or requests may be received to carry out various mechanical repairs, and were this work not so interesting it might become somewhat of a tie.

Recently we dealt with a friend's complaint that his lathe suffered badly from vibration. The lathe in question, of reputable make, was mounted on a heavy bench, and driven by an electric motor from a home-made countershaft of good design and sound construction. When the lathe was running, the vibration under some conditions seemed to build up and shake the whole bench. This vibration, however, did not depend on the speed of the lathe mandrel, and the owner had found that the trouble almost disappeared when a smaller motor pulley was fitted to reduce the countershaft speed. On removing the lathe driving belt and running the countershaft light, a marked tremor was felt on touching the bearing housings, and this became greatly increased when a rag was pressed against the cone pulley to act as a brake and slow the shaft. The cone pulley was, therefore, removed and the vibration was still present with the countershaft pulley alone rotating. The cone pulley was machined all over and so was hardly likely to be at fault; nevertheless, it was checked, with the driving pulley removed, and found to be in static balance, as it did not tend to come to rest in any one position. This test does not, however, eliminate out-of-balance forces set up when the part is rotating; that is to say, by the formation of a "couple" which may arise where two or more heavy points are spaced some distance apart along the axis of rotation. As this subject was admirably dealt with recently in THE MODEL ENGINEER, no further explanation is needed.

As the faulty, 10 in. diameter, driving pulley under consideration had a thin web and a rim only $\frac{7}{8}$ in. in width, it could be regarded as a disc, and, therefore, probably free from coupled forces tending to upset the running balance; this left only the static balance to be corrected.

Static Balance

Testing pulleys for static balance is usually carried out by mounting the pulley on a shaft supported on knife edges in order to reduce the bearing friction as much as possible. The countershaft was, however, mounted in self-aligning, ball-bearings, which were correctly fitted by clamping the two inner races to the shaft, and one outer race in its housing, leaving the second outer race a sliding fit to allow for expansion. On spinning the large pulley, alone mounted on the countershaft, it came to rest every time in almost exactly the same position; moreover, it swung to and fro until the heavy point finally settled in the lowest position. This test served to corroborate the previous findings in incriminating the pulley, and the next step was to remove some metal from the web to obtain static balance.

The Balancing Operation

The first thing to be determined is the exact position of the heavy point, so that this can be lightened by drilling, and much time and trouble may be saved by setting about the job methodically, instead of drilling holes in a haphazard manner and trusting largely to luck for a successful result. The free-running pulley was spun by hand twelve times in succession, and both the direction and force of rotation were varied. Each time the pulley came to rest, a line was drawn with a grease pencil to denote the lowest point and, from the adjudged mean centre point of the marks, a line was drawn right across the pulley through its centre, as shown in Fig. 1.

This line then passes through the points, lying on each side of the pulley centre, where weight must be either added or subtracted to effect a balance. The easiest way of adding a trial weight is, perhaps,

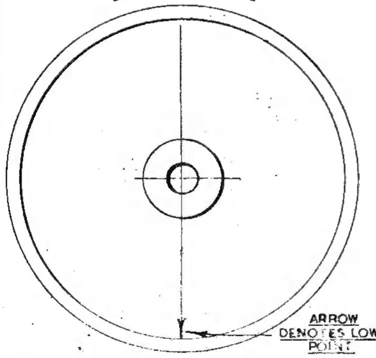


Fig. 1. Marking the heavy point on the pulley

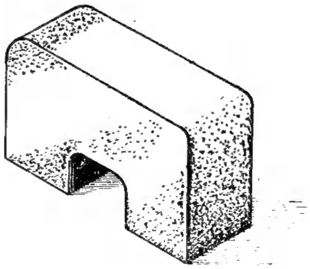


Fig. 2. A magnet used as a temporary balance-weight

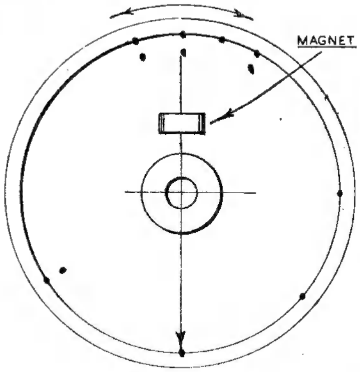


Fig. 3. Result of a test with the magnet attached opposite to the heavy point

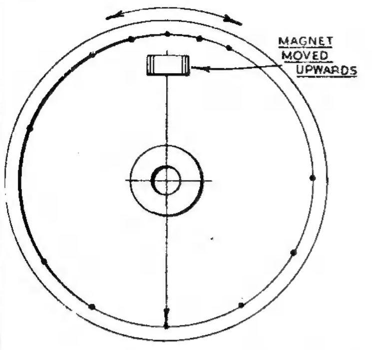


Fig. 4. A second test made after adjusting the position of the magnet

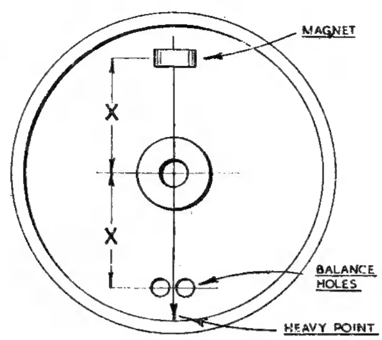


Fig. 5. Marking-out the drill holes from the position of the magnet

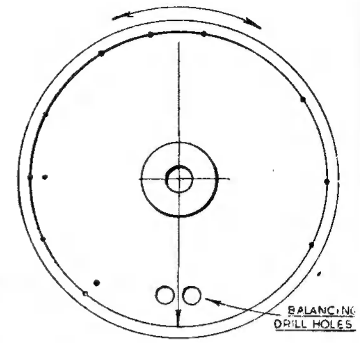


Fig. 6. Result of a final test after drilling the balance holes

by attaching a small magnet so that its position on the centre-line can be readily adjusted without in any way damaging the pulley itself. For this purpose, the Eclipse series of permanent magnets will be found convenient, and that illustrated in Fig. 2, $\frac{7}{8}$ in. in length and weighing about $\frac{1}{2}$ oz., was used in the present instance. The magnet is applied to the face of the web exactly on the scribed centre-line and on the side opposite to the heavy point.

The result of twelve trial spins is illustrated in Fig. 3, where the top of the resting pulley is, in each case, marked by a dot.

As there are seven dots in the upper half and only four in the lower, this means that the lower part of the pulley is still too heavy. The magnet was, therefore, moved upwards for a short distance and another trial made. The outcome is shown in Fig. 4, and, as the dots are now fairly evenly distributed, the pulley may be regarded as well enough balanced for all ordinary purposes.

Even were the pulley in perfect balance and running without friction, a regular distribution of the dots would not be expected with so few spins; this, of course, is also characteristic of a roulette wheel.

Drilling the Balancing Holes

It now follows that, to balance the pulley, a mass of metal equal to that of the magnet must be removed at a corresponding point on the opposite side of the centre, and this is done by drilling. Mark-out two hole centres, one on each side of the diameter line and at the same radius as that occupied by the magnet; to allow for drilling holes up to $\frac{1}{2}$ in. in diameter, the hole centres were spaced $\frac{5}{8}$ in. apart, as shown in Fig. 5.

Now, the thickness of the pulley web at this point was found to be 9/32 in. and, if the two holes are drilled right through, this will correspond to removing a bar of metal $\frac{9}{16}$ in. in length.

To determine the diameter of this bar, and so that of the drill holes, the magnet was put on a sensitive letter scales and a $\frac{9}{16}$ in. length of $\frac{7}{16}$ in. diameter steel rod placed in the other pan. As this piece of rod proved to be too light, a corresponding length of $\frac{1}{2}$ in. diameter rod was tried, and this was found to be correct with only an insignificant error.

All was now plain sailing, and two $\frac{1}{2}$ in. diameter holes were drilled at the marked-out centres. Although the result could hardly be in doubt, a further test was made to check the balance of the corrected pulley.

The result obtained from a single test, made by again spinning the pulley twelve times, is shown in Fig. 6, and it will be seen that the stopping points are remarkably evenly distributed; in fact, more regular than would be expected from these chance happenings. The final test was to try out the lathe under working conditions, but no appreciable vibration was then found with the countershaft turning at its normal speed of 400 r.p.m., and the lathe mandrel running at slow, medium, and fast speeds.

Which goes to show that a little planning at the outset often points the way to doing a satisfactory job without needless waste of time and effort.

Dividing Device for the "M.L. Super 7" lathe

(*Continued from page* 170)

screw, or the whole assembly taken away by unscrewing two Allen screws. The further details can be seen from the drawing.

The layout of the indexing detent is shown in Fig. 2 which should be self-explanatory. The lip of the back gear casing should be levelled and trued up ; to do this, remove the small rubber buffers, file the lip as flat as possible, and then scrape true, using a small piece of plate glass as a surface plate.

Next mark the position of the two holes, and drill and tap them 2 B.A. Now cut out and file up the detent plate from $\frac{1}{8}$-in. mild-steel ; drill the hole and cut the slot, and the job is done, except for filing up the engaging edge to an involute shape to suit the tooth gap.

To operate, the belt case is lifted, the clamp-screw is loosened, and the detent plate pushed into engagement with the bull wheel by means of the knob on the plate. If heavy work is to be carried out, such as filing a square on the end of a shaft, then it would be as well to reclamp the detent plate before carrying out the work.

The teeth were numbered in the following way : The bull wheel was first given a coat of cellulose primer, and then a coat of cellulose varnish. The figures were then printed in ink on a piece of thin cartridge paper, and the circle cut out and also cut in half. The back of the paper was given a coat of varnish, and whilst wet, applied to the wheel, where it stuck firmly. When dry, two coats of varnish were applied to the front of the paper, making it impervious to oil and grease—I hope.

West Midlands Federation Cup Competition

Reported by "Northerner"

THE Cup Competition of the West Midlands Federation of Model Engineering Societies naturally attracts the cream of the models in that area, and this year's event was no exception. There were only ten entries, but they were all of an extremely high standard, and the judges had a difficult task.

However, most of the work displayed has already been described, and some of the entries illustrated, in my report on the Birmingham Exhibition, so that there is not much to say here. Winner of the chief trophy, the West Midlands Federation Cup, was the very fine ex-American W.D. 0-6-0 locomotive, built in 5-in. gauge by J. Strickland of the Rednal Club.

This is an impressive-looking locomotive, as will be realised from the photograph which appeared recently, and we would have liked to see how she performs on the track.

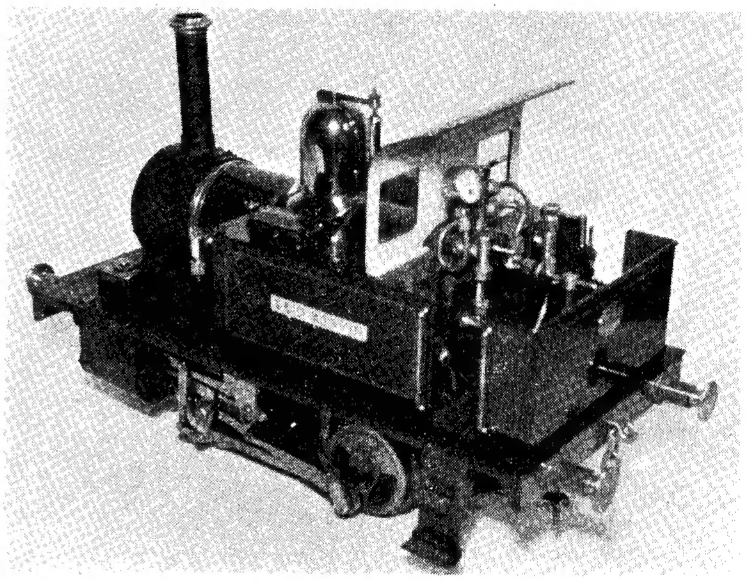
A ship-model was awarded the second prize, the Addenbrooke Cup; this was A. E. Field's early 16th-century Spanish Carrack built to 1/60th scale.

The model is based on that in the Science Museum, which is a copy, presented by the Spanish Government, of the model of Columbus's flagship *Santa Maria* built in Madrid in 1892. However, since that date numerous anachronisms have been detected, and Mr. Field has made certain alterations of his own, so that his model has the title stated above.

The runner-up was the lovely little $3\frac{1}{2}$-in. gauge 0-4-0 contractors' locomotive, *Maid Marion*, built by J. H. Balleny, of the Birmingham Society. This model has not been illustrated previously, so here is a picture of her.

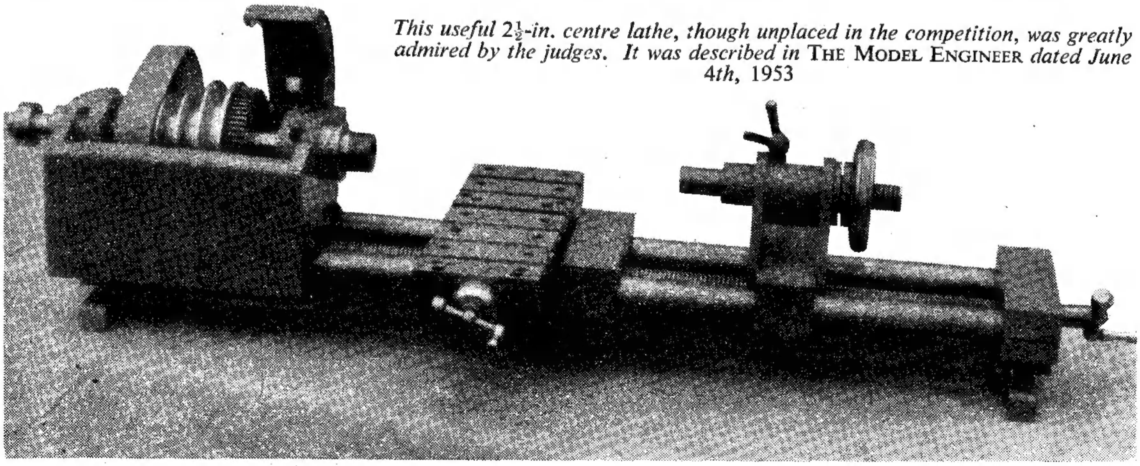
Another entry was the $2\frac{1}{2}$-in. centres lathe built by J. Harbidge, Junior, of Marston Green—a really excellent piece of work for a youth of sixteen.

The competition was combined with a rally at the Birmingham Society's ground at Sheldon, and although the attendance was not very heavy, a really good time was had by all, including yours truly. But what better way could one find of spending a summer afternoon, with some good models to look at, a bunch of good chaps to natter with, and a few rides round the track behind a good locomotive?



"L.B.S.C.'s" popular "Tich," with embellishments, resulted in J. H. Balleny's "Maid Marian"

This useful $2\frac{1}{2}$-in. centre lathe, though unplaced in the competition, was greatly admired by the judges. It was described in THE MODEL ENGINEER *dated June 4th,* 1953



QUERIES AND REPLIES

" THE M.E." FREE ADVICE SERVICE. Queries from readers on matters connected with model engineering are replied to by post as promptly as possible. If considered of general interest the query and reply may also be published on this page. The following rules must, however, be complied with:

(1) Queries must be of a practical nature on subjects within the scope of this journal.
(2) Only queries which admit of a reasonably brief reply can be dealt with.
(3) Queries should not be sent under the same cover as any other communication.
(4) Queries involving the buying, selling, or valuation of models or equipment, or hypothetical queries such as examination questions, cannot be answered.
(5) A stamped addressed envelope must accompany each query.
(6) Envelopes must be marked " Query " and be addressed to THE MODEL ENGINEER, 19-20, Noel Street, London, W.1.

Watercooling Petrol Engine

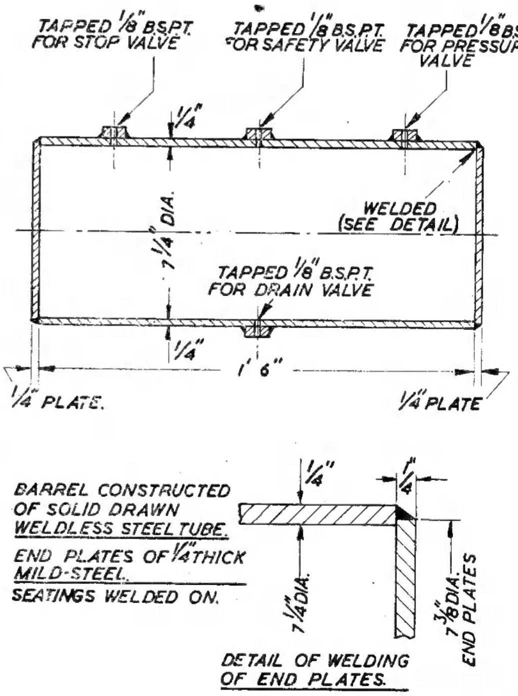
I have a small petrol engine fitted to a boat, but find that it gets much too hot after running for a minute or two. I have thought of watercooling it, by machining away the fins and fitting a brass tube on the cylinder to serve as a water jacket, but I am told that it will be necessary to fit a jacket to the head as well. As this is much more difficult to carry out, I shall be glad of your advice on the matter. Also, will it be necessary to use a pump to keep up the circulation of water through the jacket? What type and size of pump do you recommend? B.H. (Coventry).

Theoretically, the whole of the cylinder exposed to internal heat should be jacketed, and if there is any discrimination in the matter, the head is more likely to require cooling than the barrel. But in practice, small engines may be effectively cooled by jacketing the barrel only, as the conduction of heat from the barrel to the head is very rapid. It would be necessary to modify the head design considerably to enable it to be jacketed. Examples of cylinders with (a) barrel only jacketed and (b) barrel and head jacketed will be found in our handbook, *Model Petrol Engines*, price 7s. 6d.

Some form of forced water circulation is necessary unless the engine jacket system is submerged below the water level, which is unlikely in a normal boat installation; or alternatively, a closed circuit water system, with a header tank or radiator inside the boat, may be employed, if the weight is not prohibitive. A circulating pump is desirable, but the same purpose can be served by utilising the forward motion of the boat to drive water into a scoop facing forward, and leading as directly as possible to the water jacket; this system, however, is operative only when the boat is actually under way, and fails to cool the engine if the boat is held stationary any length of time with the engine running. Any type of pump may be used, but small rotary pumps of the centrifugal, vane, or gearwheel type are simplest to make and install. Quite a small pump will supply sufficient water for cooling; as an example, a pump with an impeller $\frac{3}{4}$ in. diameter, running at engine speed, has been found effective in the case of a 15 c.c. engine.

Air Compressor

I have a small air-cooled air compressor of $1\frac{1}{2}$ in. bore by $1\frac{1}{2}$ in. stroke, which I propose to use for the purpose of paint spraying, and brazing with coal gas. I propose to drive this with a $\frac{1}{4}$ h.p. single-phase 240-volt motor, running at about 1,400 r.p.m. Will this motor drive a compressor and enable it to maintain a pressure of 60 lb. per sq. in.? At what speed should I run the compressor? Is it large enough to maintain the pressure for the above-mentioned purposes? Will the air receiver shown in the sketch be safe for 60 lb. per sq. in pressure? W.R. (Kilmarnock).



The motor you propose to use for your small air compressor should be quite satisfactory, and a suitable speed for the air compressor would be about half motor speed, so that if driven by a Vee belt, which would be a very suitable method of driving, the pulley on the compressor shaft, should be twice the size of that on the motor.

This plant will not supply sufficient air for continuously working a normal full size spray gun, but it can be used very satisfactorily for intermittent work, particularly with a miniature spray gun, such as the " M.E." type.

Most spray painting can be done at much lower pressure than 60 lb. per sq. in., which we think would be difficult to maintain with this plant.

With reference to the proposed air receiver, this should be of ample strength if it is properly constructed, but in the case of any pressure vessel we recommend that it should be tested by hydraulic pressure before putting it into service.

Incidentally, a longitudinal stay through the centre of the receiver would very much increase its strength, as the ends of a pressure vessel are usually the weakest.

Electric Hedge Cutter

Will you please inform me of the date of the issue or issues of THE MODEL ENGINEER *containing an article on the construction of an electrically-driven hedge cutter?* W.B.C. (Manchester).

This article appeared in the issue of THE MODEL ENGINEER dated January 19th, 1950, a copy of which may be obtained from our publishing department, price $10\frac{1}{2}$d., post free.